PRODUCT ENGINEERING

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The Patent Conflict

EVERY reform movement presents the possibility of committing two grave errors. Sweeping conclusions are too often drawn from specific cases. Secondly, the cure might be worse than the disease. But in the case of the present "patent battles" another question arises, namely, it is doubtful if some of the practices now frowned upon by the Department of Justice are actually evils that conflict with the public welfare.

According to a statement credited to the Department of Justice, "Patent rights have been used to divide fields into non-competitive spheres . . . to restrain trade and eliminate competition." This presumably refers to the general practice of issuing a license to a manufacturer which permits him to apply that patent only to certain specified products. Presumably in the eyes of the Department of Justice this is an evil because it "restrains trade" and "restricts output."

As an example, let us assume that the Packard Motor Car Company or the Studebaker Corporation invented a wonderful engine of extraordinary power and efficiency, and that patents were procured. According to the viewpoint expressed above, it would be contrary to the public welfare if, in this assumed case, Packard or Studebaker were to license General Motors to manufacture these engines, but only for their application to aircraft, and definitely not for their application to passenger automobiles. It is readily apparent that probably the best procedure in this hypothetical case would be for the owners of the patent not to license anyone as they would be inviting competition in the application of this new engine to automobiles. Under such conditions the aircraft industry would be deprived of a far superior engine.

Another fact which the Department of Justice apparently contends is an evil in restraint of trade and which eliminates competition is the issuance of patent licenses wherein limited production and quality are specified. Although at first glance it might appear to be contrary to the public welfare to limit production and quality on a patented product, there are instances on record wherein specified provisions covering production and quality were incorporated in the licensing agreement to the undoubted benefit of public welfare. Regardless of the values and virtues of a new invention, if the machine or device as manufactured is of inferior quality and workmanship, it will not find public acceptance. Therefore, it is often necessary for the inventor to incorporate in his license agreement certain provisions that will require the manufacturer to maintain or exceed a specified minimum standard of high quality. This serves to protect the public against inferior products and the public is benefited accordingly.

With reference to limitations on production, it is generally admitted that unbridled competition and production often ruins an industry. In the long run it may be to the public welfare for the owner of a patent to license certain manufacturers to make a limited number of units per year rather than giving a number of manufacturers unrestricted licenses. The government maintains that production control is desirable in agriculture.

Other so-called evils frowned upon by the Department of Justice include the division of markets into exclusive geographical areas and the restriction of the use of the product to certain definite applications.

It is generally feared that if the Department of Justice were to succeed in getting Congress to pass laws which will legally put an end to all the practices upon which it frowns, the term patent protection will have little or no significance or value. On the other hand, there is probably some room for improvement in the present laws. In order to inform the readers of PRODUCT ENGINEERING concerning the present status of patent laws and the proposed changes, we have commissioned Karl Fenning, the well-known Washington patent lawyer who was formerly Assistant Commissioner of Patents and is now editor of *U. S. Patents Quarterly*, to present the complete picture of the patent situation in PRODUCT ENGINEERING. His first article, to appear in the February number will cover the present status of patent laws and his second article will discuss proposed changes.

CHAIN DRIVE CALCULATIONS

A New Design Analysis

IN the calculation of chain drives according to the method approved by the American Standards Association, it is assumed that when the chain links are in mesh with the rotating sprocket, the centrifugal force developed is transmitted throughout the length of the chain. Quoting from an article which appeared in the May, 1935 number of PRODUCT ENGINEERING:

"Further in connection with chain speeds is the question of possible effects because of centrifugal force. The tension created by centrifugal force can be calculated by the equation:

$$\text{Chain Centrifugal Load} = \frac{\text{wt. per ft. of chain} \times V^2}{115{,}900}$$

"This equation gives the load in pounds developed by the centrifugal force at a chain velocity of V ft. per min., and this load must be added to the horsepower load.

"The above equation may be recognized as being derived from the well-known formula for the stress created in a flywheel rim by centrifugal force. In this connection it is also interesting to consider the formula for the calculation of permissible load as approved by the American Standard Association, namely:

$$T = \frac{2{,}600{,}000A}{V + 600} - \frac{WV^2}{115{,}900}$$

"In this equation T is given as the working load in pounds, A is the projected pin bearing area in square inches, V is the chain velocity of feet per minute and W is the weight per foot of chain. . . .

"Centrifugal force is not a factor in establishing the rating of chain to be used because the published ratings (published by the chain manufacturers) include allowances for centrifugal force. Therefore, centrifugal force should not be added to the calculated working load on either roller or silent chain.

"For most high-speed applications a chain speed between 1,500 and 2,800 ft. per min. is the best, but much faster or slower drives can be designed to operate equally well."

In the May 1935 number of PRODUCT ENGINEERING the author referred to the "question of possible effect because of centrifugal force." The inclusion of the word "possible" was perhaps more or less in the nature of an accident or merely a reflection of the ultra-conservativeness of an engineer. But, regardless of any apologies or explanations, the fact now has been proved definitely that the centrifugal force developed when the chain is traveling through its arc of

Fig. 1—The centrifugal force developed is balanced by the components of the reactions of the teeth against the rollers. Each of the two teeth exerts a vertical force equal to one-half the centrifugal force

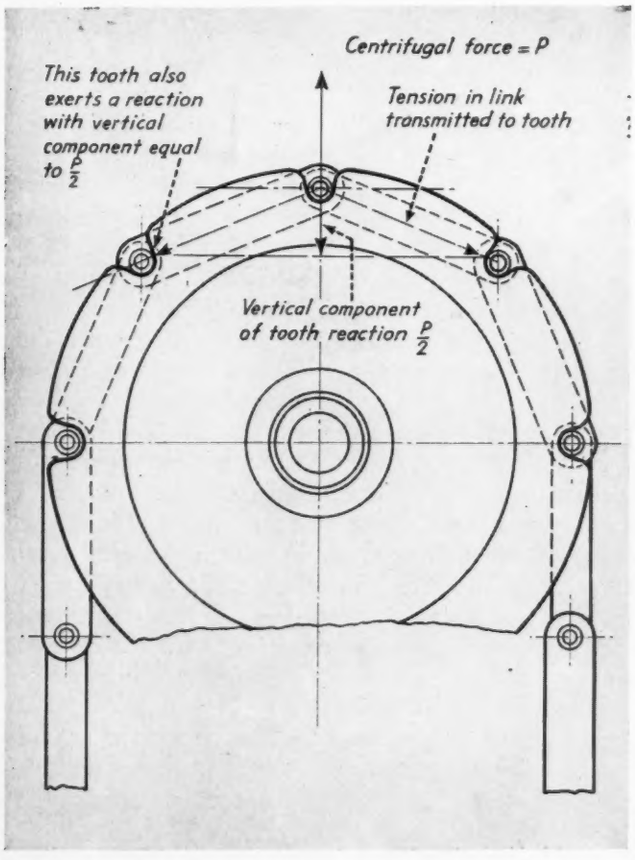
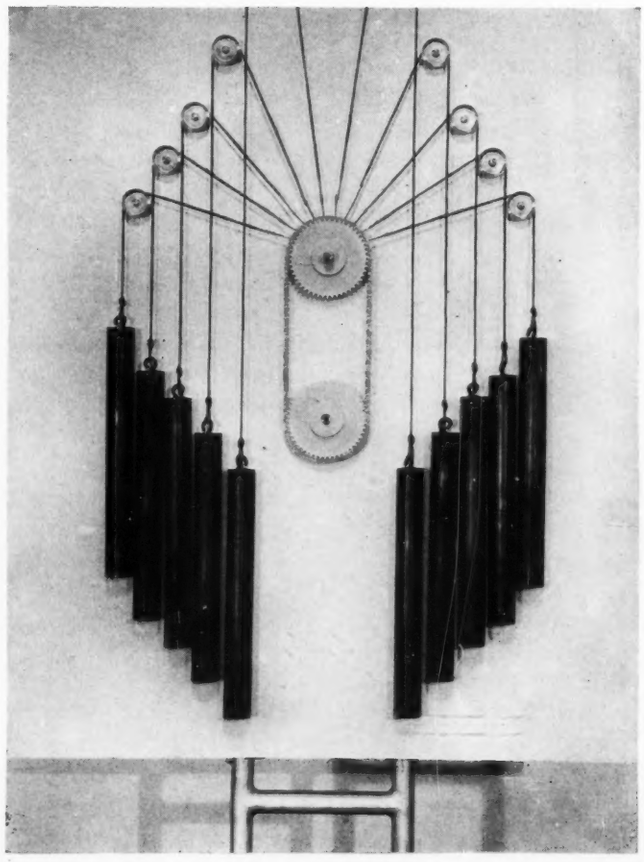

Centrifugal force = P
This tooth also exerts a reaction with vertical component equal to P/2
Tension in link transmitted to tooth
Vertical component of tooth reaction P/2

Fig. 2a—The upper sprocket is mounted on a shaft. The lower sprocket has no shaft; it merely hangs in the chain. The weights simulate centrifugal forces and are balanced by the reactions against the teeth



contact with the sprocket wheel, is not transmitted to either of the straight portions. The centrifugal forces are practically all transmitted to the teeth of the sprockets; the analogy between centrifugal forces developed in the rim of a flywheel and those developed in a chain, does not apply. This is not a mere theory, but is a fact that has been demonstrated by the engineers of the Morse Chain Company and is proved by not only logical analysis, but also by tests, experiments and the records of actual applications.

The long-standing erroneous theory concerning the centrifugal force that is developed when a chain rotates with the sprocket wheel resulted from the assumption that the action is the same as that produced in the rotating rim of a flywheel or when a flat belt goes around a rotating pulley. Practically the only similarity between the rotating rim of the flywheel and a chain being carried around on its sprocket is that in both instances centrifugal forces must of necessity be created. However, the rim of a flywheel is held only by the radial spokes of the wheel and therefore the centrifugal forces must be transmitted through the rim to the spokes. But in the case of chains, the chain links are held between the teeth of the sprocket. This results in an entirely different situation.

In order to simplify the analysis of the action of centrifugal force on roller or silent chains, let it be assumed that a heavy weight is fastened to one of the links of the silent or roller chain and that the chain itself is weightless. A full explanation of what occurs is given in the diagrams shown in Fig. 1. As noted in the diagrams, practically none of the centrifugal force developed can be transmitted to the straight portion of the chain drive. The faces of the sprocket teeth react against the chain links with a force equal and opposite to the component of the centrifugal force lying in the direction of the chordal portion of the chain.

The correctness of this theory is demonstrated by a model constructed by Mr. N. C. Bremer, chief engineer of the Morse Chain Company, and illustrated in Fig. 2. The caption and notation explain the model. The theory is also proved by tests conducted at the Morse Chain Company plant and by drives that have been running two years or more.

The tremendous significance of this discovery made by the engineers of the Morse Chain Company does not become fully apparent until a complete analysis of its effects upon the calculation of the permissible working load has been made. Of course, the last term in the above erroneous formula, approved by the American Standards Association for calculating the permissible working load for chain drives should evidently be omitted for all speeds under all conditions, because centrifugal force does not add to the tension in the chain. This increases the value of T, the effective pull, as calculated by the American Standards Association formula, by an amount equal to the pounds centrifugal force as calculated. But this is only one of the factors to be considered. Perhaps of greater importance is the fact that on the basis of this new and corrected design method, much greater chain speeds are permissible.

As was shown in Fig. 1, the centrifugal force developed must be transmitted through the chain links and thus to the working faces of the sprocket teeth. This fact establishes the limiting speed for chain drives. On this new basis of calculation, the maximum chain speed will be that at which the centrifugal load developed equals the allowable chain pull. The allowable pull is usually taken as the tensile strength of the chain di-

Fig. 2b—With three teeth on each side of the upper sprocket ground off, the tooth reactions do not balance the centrifugal forces. Hence a balancing pull of 5 lb. plus the 1½ lb. weight of the scale is needed

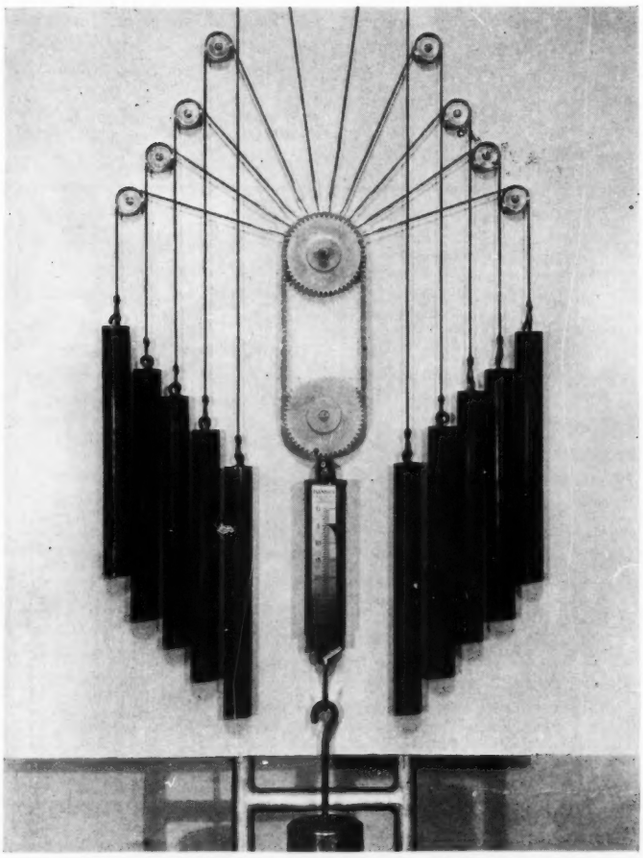
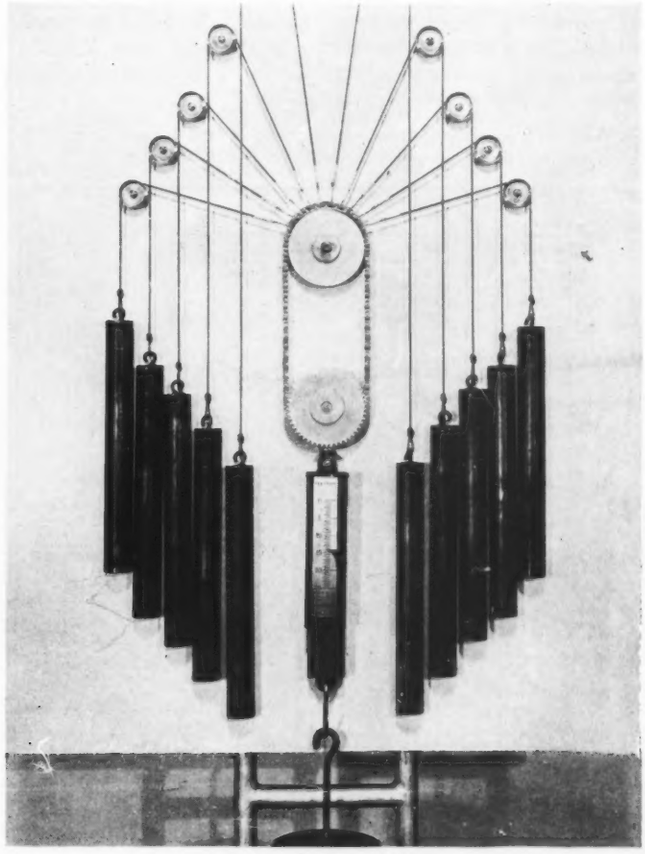


Fig. 2c—With 12 teeth of the upper sprocket ground off, it requires a total pull of 16½ lb. to balance the centrifugal forces. With all teeth ground off, balancing force must equal total centrifugal load



vided by a factor of safety of 20. For example, a 1-inch pitch Morse silent chain 1-inch wide, has a tensile strength of 15,000 lb. and weighs 2 lb. per ft. Dividing the tensile strength of 15,000 lb. by a factor of safety of 20 gives 750 lb. as the allowable centrifugal load. Using the equation given above:

$$\frac{WV^2}{116{,}000} = \text{centrifugal load} = 750 \text{ lb.}$$

For a value of $W = 2$ lb. per ft., the value of V as obtained in this equation will be 6,600 ft. per min. On this basis, about 6,000 ft. per min. is the maximum permissible chain speed, although this figure will vary according to the weight per foot of chain and its tensile strength. The safe limiting speed for various chain drives is given in the accompanying chart.

It will be observed that the centrifugal tension is not developed in the chain until the chain has wrapped on the sprocket. Also, the centrifugal tension in the chain disappears as the chain leaves the sprocket. And whatever centrifugal tension is developed in the chain is absorbed by the reaction of the sprocket teeth. Thus at the moment when there is rotation on the pin bearing, that is, when the link is going through the angle of swing or articulation which occurs when the link enters or leaves the sprocket, there is practically no bearing pressure on the pin created by centrifugal force. Hence the centrifugal force has practically no effect either on joint wear or linkage fatigue. The significance of this is that increasing the chain speed up to the limit as set forth above does not increase wear in the joints of the chain nor does it increase fatigue.

On this new basis of design of chain drives, for a given number of revolutions per minute it is now perfectly proper to increase the number of teeth in the sprocket to a point where the chain speed is 4,000 to 5,000 ft. per min. instead of the previous conventional limit of about 2,000 ft. per min. But the greater the number of teeth in the sprocket, the less becomes the "angle of swing or articulation." The smaller this angle is, the less becomes the impact developed when the chain teeth swing upon the sprocket or leave the sprocket, the smaller is the wear on the chain pins, the smoother is the running of the chain. Carrying the analysis still further, for a given horsepower the higher the chain speed the less will be the unit pressure on the projected area of the pins of the chain. For example, at 1,000 ft. per min. the allowable unit pressure on the projected area of the pins is 1,700 lb. per sq. in., while at 2,000 ft. per min. chain speed, this allowable unit pressure has not reduced to one-half, but rather only to 1,000 lb. per sq. in. Of course, these figures for allowable unit pressure on the projected pin-bearing are empirical, but are based on experimental results.

In order to illustrate the great significance of this new and proved method for calculating chain drives, comparative results as obtained by the old method and the new method will be illustrated. On page 219 of the June 1935 number of PRODUCT ENGINEERING appears the horsepower rating chart for Morse Silent Chain one inch wide. This chart, of course, assumes that the centrifugal force developed is transmitted throughout the length of the chain.

According to this chart, Fig. 3, a 1-in. pitch chain, one inch wide running on a 19 tooth sprocket at 1,000 r.p.m. can transmit 12.6 hp. The angle of articulation will be 19 degrees.

In Fig. 4 is shown the new chart for the calculation of silent chain drives. This chart has been calculated on the basis that the centrifugal force developed when the chain runs around on the sprocket is not transmitted to the straight or load carrying portion of the chain. According to this chart a one-inch pitch silent chain, one-inch wide, running over a 55 tooth sprocket at 1,000 r.p.m., at a chain speed of 4,583 ft. per min., can transmit 42 hp. The angle of articulation will be 6½ deg. or about 1/3 of that for the 19-tooth sprocket.

It will be observed that the increased chain speed is entirely the result of using a greater number of sprocket teeth. This in turn decreases the angle of articulation to about 1/3, which results in a corresponding decrease in impact, less wear on the pin bearings of the chain, smoother action and practically the elimination of noise. In other words, the advantages of the greater number of sprocket teeth is cumulative not only because the higher chain speed means a lesser chain pull required to transmit the same horsepower, but also because the running conditions, with reference to impact, and noise, are cor-

Fig. 3—Chart for calculating silent chain drives according to the American Standards Association method which is in error

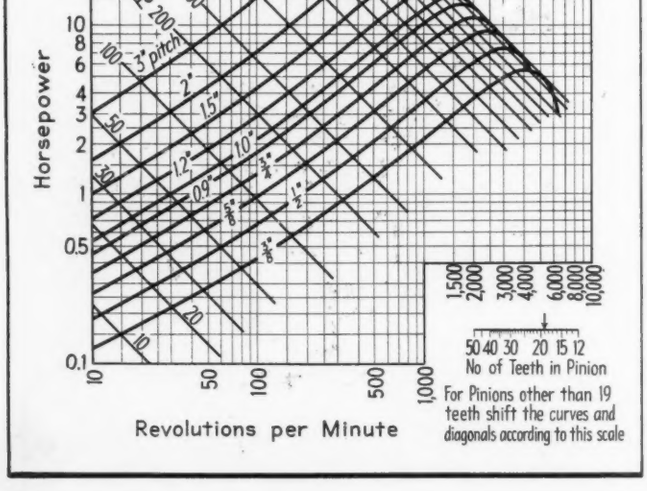
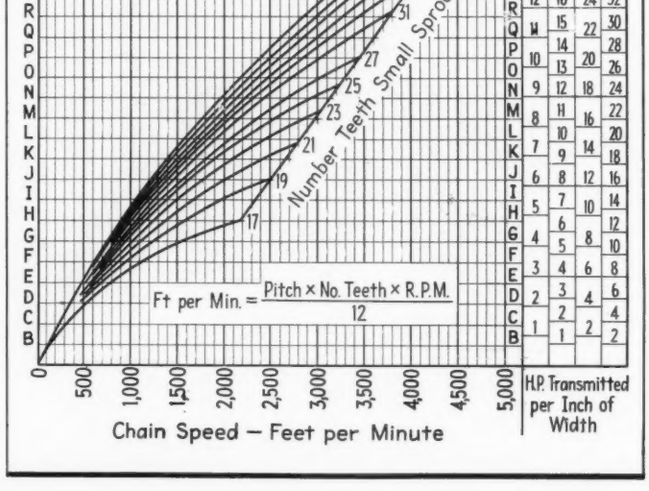


Fig 4—New chart for silent chain drives, based on centrifugal force being balanced by the tooth reactions



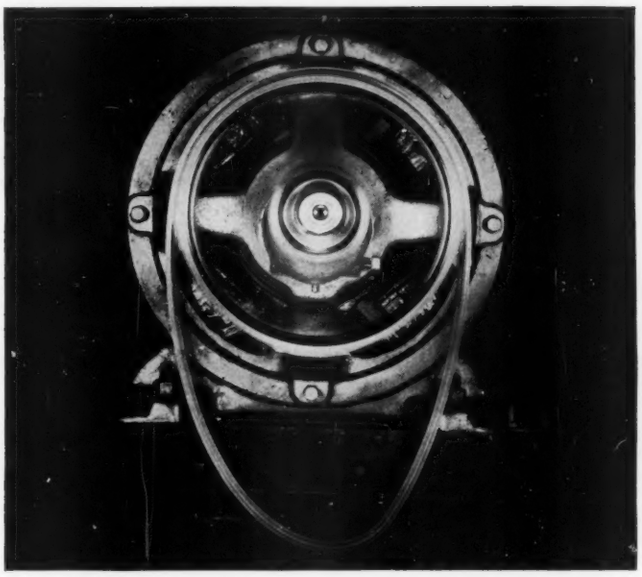


Fig. 5—If centrifugal force were transmitted throughout, this chain running 5,000 ft. per min. would fly off the sprocket

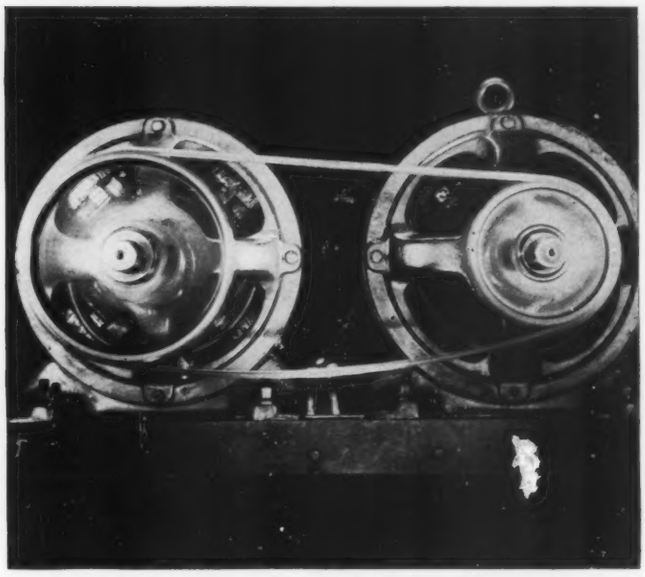


Fig. 6—This drive in the Morse Chain Co. laboratory has been running under load at 5,000 ft. per min. for more than two years

respondingly improved. And the horsepower transmitted has been increased from 12.5 hp. to 42 hp., the chain pitch and width being the same.

So startling are the results of the application of this new method for the design of chain drives that the engineers of the Morse Chain Company hesitated to announce it until the validity of this analysis had been proved by long time service in actual applications in addition to searching laboratory tests.

A number of long-time laboratory tests were made. In one of these a 50 hp. d.c. motor drove a generator at 1,670 r.p.m. The number of teeth in the driver sprocket was 93 and the number of teeth in the driven was 65, giving a chain speed of 5,650 ft. per min. and a working load on the chain of 300 lb. The silent chain was $\frac{5}{8}$ in. in pitch by 1 in. wide and according to calculations developed a centrifugal tension of 345 lb. This drive has been running 8 hr. per day for 19 months at the end of which time the chain was remeasured and showed a chain wear of only 0.00055 in. per pitch. The soft iron sprockets showed only slight wear. If the sprockets had been flame hardened Meehanite the wear would have been practically zero.

Among the industrial applications is a 60-hp. diesel engine running at 1,500 r.p.m. driving a hammer mill at 3,200 r.p.m. The driving pinion had 71 teeth and the driven sprocket on the hammer mill, 33 teeth. Chain speed is 4,440 ft. per min. and the working load is 450 lb., chain pitch is $\frac{1}{2}$ in. After five months operation at 20 hr. per day the chain and sprocket were still in excellent condition and no adjustment of the centers was necessary.

Among many other installations are 40, 50 and 120 hp. drives operating at chain speeds ranging from 4,000 to 5,000 ft. per min. In every instance these drives have not only operated successfully, but have shown no appreciable wear after months of service.

A number of factors must be observed in the design of these high speed chain drives. The recommended pinion sizes for different speed ratios are given in the following table:

1 to 1 speed ratio—65 teeth
2 to 1 speed ratio—50 teeth
3 to 1 speed ratio—43 teeth
4 to 1 speed ratio—35 teeth

For 1 to 1 speed ratio, a chain speed of 5,500 ft. per min. should be aimed at, this chain speed being slightly reduced as the speed ratio increases. For a 4 to 1 speed ratio a chain speed of 4,500 ft. per min. is recommended. The practical limitation that is imposed by the high-speed chain drive is the fact that when the speed ratio gets too large the sprockets become too large, and the chain speed should not exceed 5,500 ft. per min., although drives at 6,000 ft. per min. have operated successfully.

The minimum number of teeth to be used in high-speed chain drives is generally taken as 35 teeth and the pitch of the drive is selected so as to give the desired feet per minute chain velocity. That is, the pitch is determined by the r.p.m. of the lower-speed shaft. For silent chain drives, the necessary width of the chain is calculated after the pitch has been selected. In the case of roller chains, multiple strands are used, as may be required.

With reference to the efficiency, the only appreciable losses in these high speed chain drives are the windage losses. Long time tests on drives with 17 tooth sprockets have given overall efficiencies of 98.6 per cent, these efficiencies including the bearing losses. The same drive with 53 tooth sprocket gave an over all efficiency as high as 99.4 per cent.

In the installation of high speed silent or roller chain drives the chain is installed with a minimum of operating slack. It is customary to install so that with the drive stationary, both strands of the chain are fairly taut. When the drive is operating the normal chain elasticity will assure proper slack on the unloaded side.

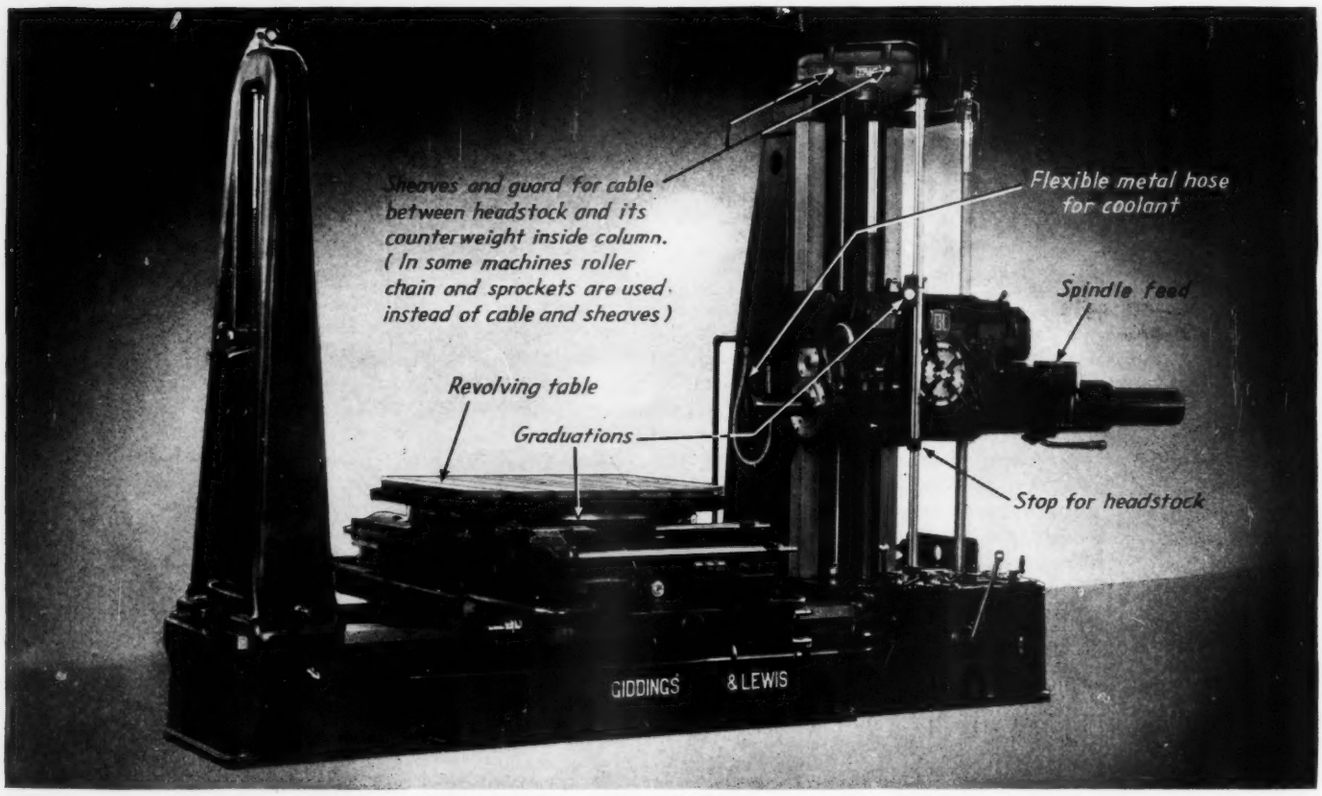
For best lubrication an automatic pump is used, although many drives give satisfactory service when the lower side of the chain or the lowest point in the chain drive runs through oil, the level of the oil in the case coming just high enough to touch the chain. S.A.E. 20 or S.A.E. 30 oil can be used, but whatever oil is used must be a light oil and one that will by capillary action penetrate to the pin bearing areas.

To minimize wear on the teeth Meehanite flame-hardened sprockets are used. It has been found by experience that with such sprockets the wear on the sprocket teeth is negligible.

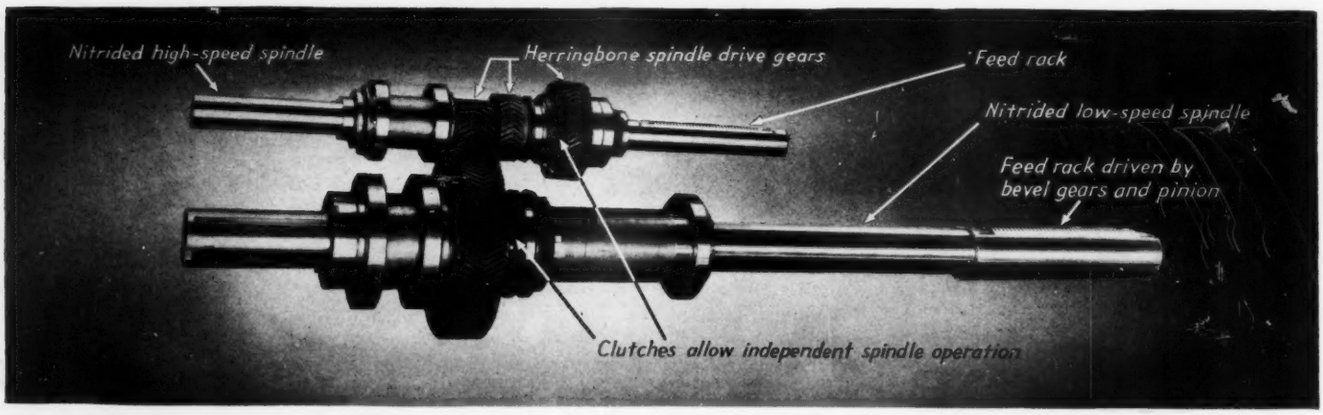
Although the above discussion has been confined mainly to silent chain drives, the same design procedure applies to roller chain drives. The revised chart for the capacity of Morse roller chain drives is shown in the accompanying figures, these capacity curves being constructed in the same manner as those for the silent chain drives. The only difference in the application of these roller chain curves is that multi-strand roller chain is used, whereas in the case of silent chain drives, the width of the chain is selected to give the horsepower capacity desired.

Modern Designs

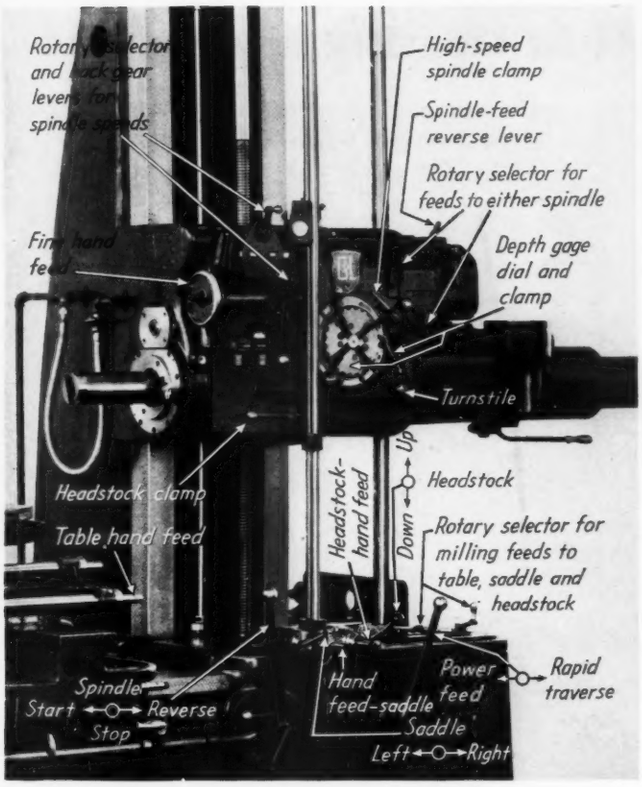
Revolving Table in Horizontal Boring Machine


Sheaves and guard for cable between headstock and its counterweight inside column. (In some machines roller chain and sprockets are used instead of cable and sheaves)
Flexible metal hose for coolant
Spindle feed
Revolving table
Graduations
Stop for headstock
GIDDINGS & LEWIS

Beds, columns, tables and saddles in this Giddings & Lewis machine are of close grain iron alloy of about 45,000 lb. per sq.in. tensile strength. Headstock is lubricated by built-in gear pump which cascades oil over moving parts. Gears in bed and in table-drive-unit are lubricated by splash system. Bijur force-feed system meters filtered oil to table and saddle ways. Bearings on counterbalance sheaves and headstock-drive shaft are grease-cup lubricated.

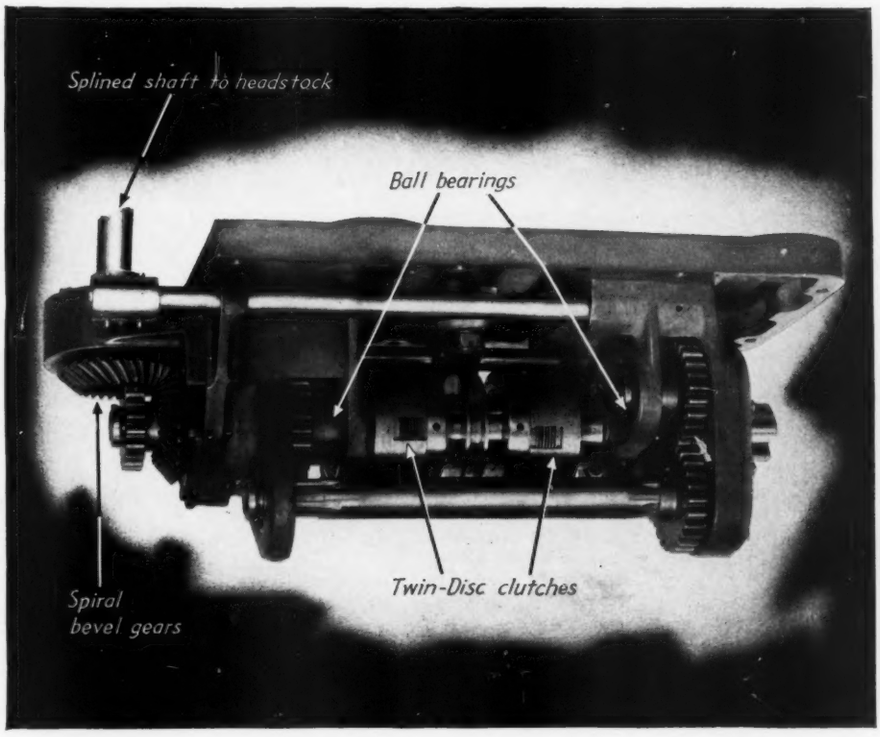

Nitrided high-speed spindle
Herringbone spindle drive gears
Feed rack
Nitrided low-speed spindle
Feed rack driven by bevel gears and pinion
Clutches allow independent spindle operation

Two Nitralloy spindles are used in headstock, the larger for speeds up to 450 r.p.m., the smaller for speeds up to 1,500 r.p.m. On the larger, sleeves and bearings can be de-clutched to eliminate fly-wheel effect on the small spindle. Spindles are driven through heat-treated, alloy-steel, drop-forged gears. Main spindle bushings are hardened S.A.E. 52100; main spindle sleeve is a heat-treated, nickel-alloy forging. Both spindles are mounted in preloaded bearings.

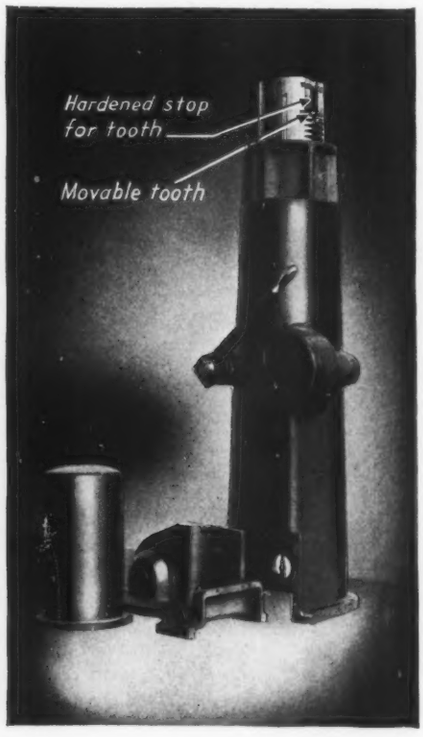


All gear train movements are controlled by levers centrally located at front of machine. Those controlling spindle feed, table, saddle and headstock are arranged to move in same direction as the desired motion of the part being moved. No keys are used, all shafts and clutches being splined.

Standard, constant-speed motor mounted on a hinged bracket on end of machine drives gear trains through Tex-rope drive. Power is transmitted to headstock through vertical splined shaft which is of ground, high-carbon steel. Short spline shafts are heat-treated alloy steel.

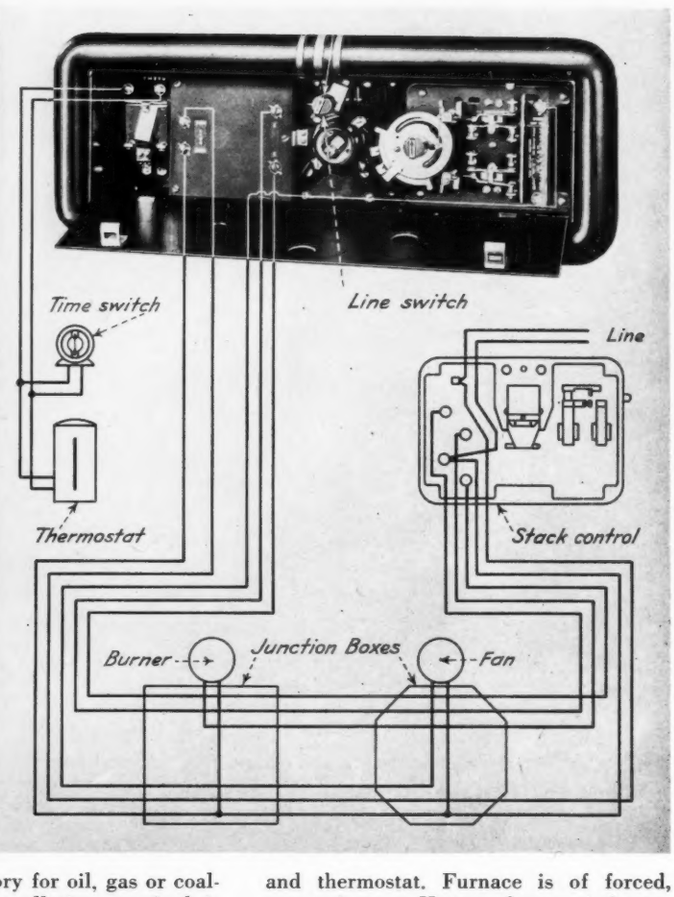


Spindle rotation is reversed by unit located in bed of machine and carrying two Twin-Disc clutch units. Separate gear trains provide 36 spindle speeds; 18 independent milling feeds to table, saddle and headstock; 18 independent feeds to either spindle. Gears are heat-treated, chrome-moly steel.

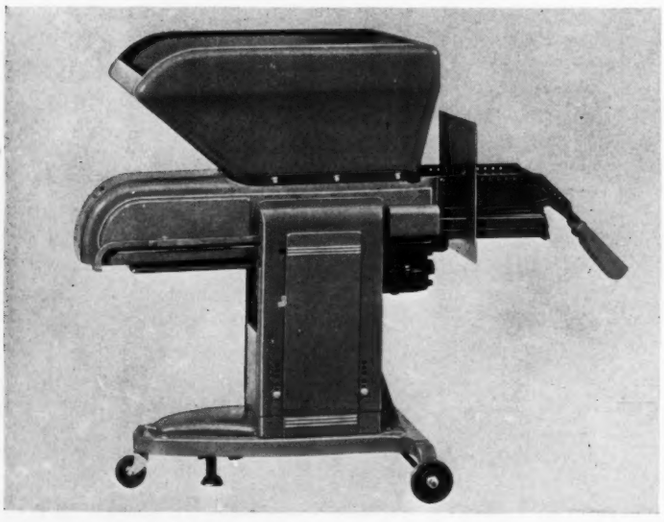


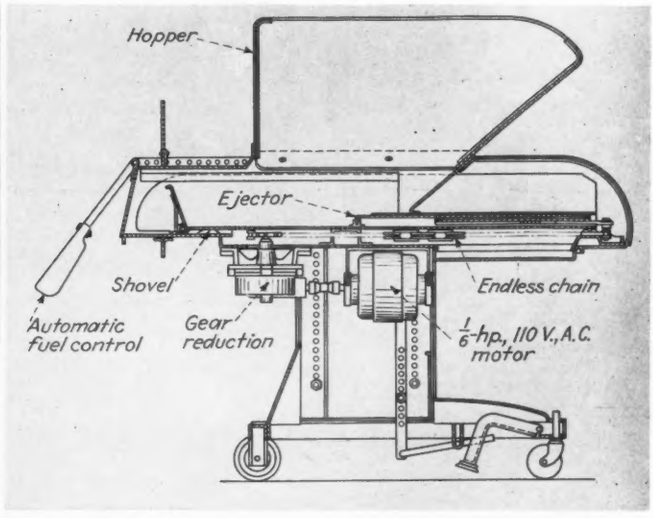
Ram safety device. Last tooth in spindle ram, held in place by spring, acts as ratchet. When bevel gear driven pinion strikes it, tooth moves and ram stops.

MODERN DESIGNS — New Trends in Heating Equipment



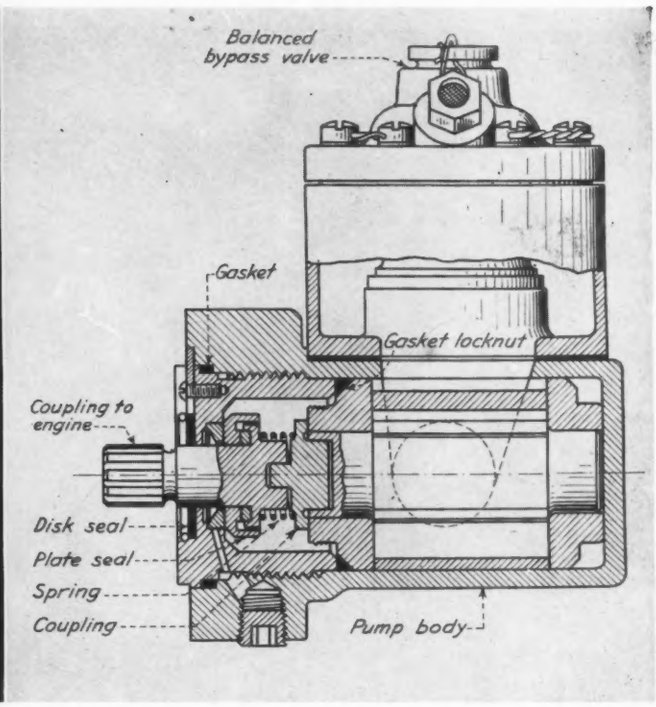
Pre-fabricated wiring harness, which reduces labor cost in installation materially, is standard on the new Schwitzer-Cummins warm air furnaces. Panel is furnished complete with controls wired at factory for oil, gas or coal-firing; the only installation required is to fasten the rubber molding around control box over six grommets and connect the proper wires to supply line and thermostat. Furnace is of forced, warm-air type. Heat exchanger is heavy-gage steel, welded. Seven-gage steel is used in the combustion chamber, ten-gage in radiator section.



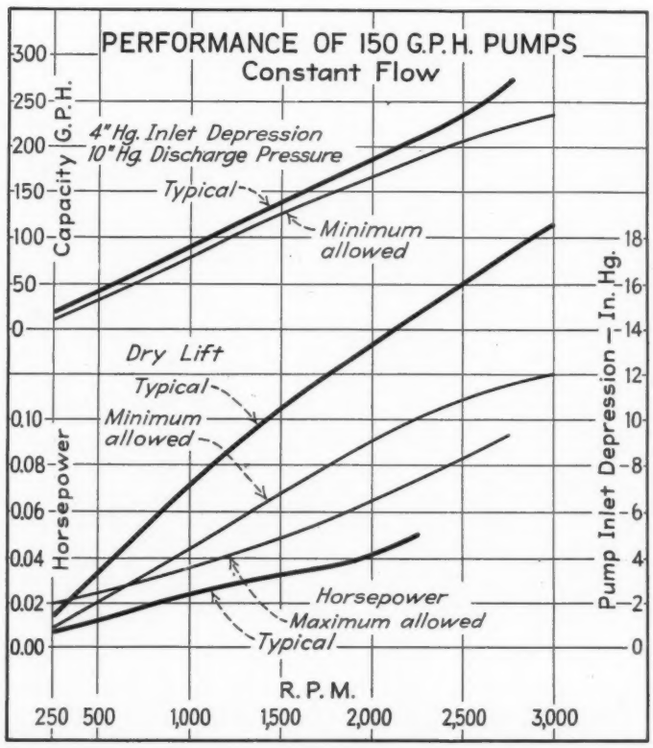


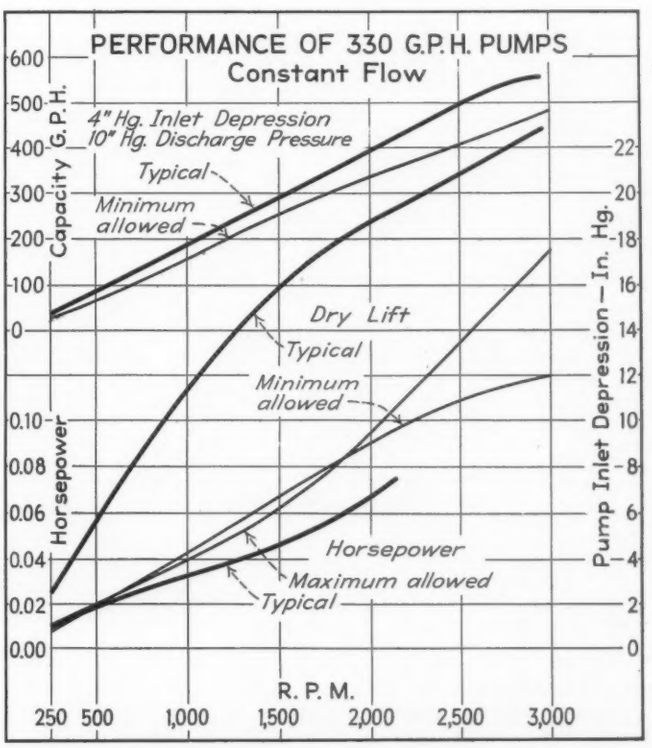
Mounted on hard rubber casters, the Murray stoker can be wheeled from furnace to bin for refilling. Plugging-in to electrical outlet is the only installation required. Height can be adjusted to fit furnace door; stroke can be adjusted to suit firepot diameters from 16 to 34 in. Automatic fuel control has a mercury switch which closes motor circuit to start injector when control arm drops as fuel changes to ash. Base, body and hopper are steel stampings. Injector drive consists of a 1/6 hp. rubber-mounted motor, self-lubricated gear reduction unit and hardened roller chain.

Non-Pulsating Aircraft Fuel Pumps

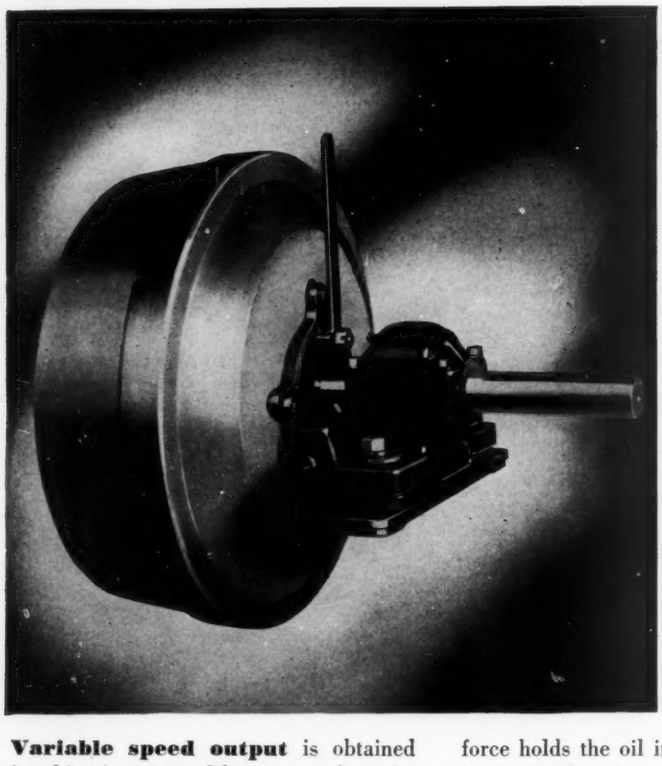


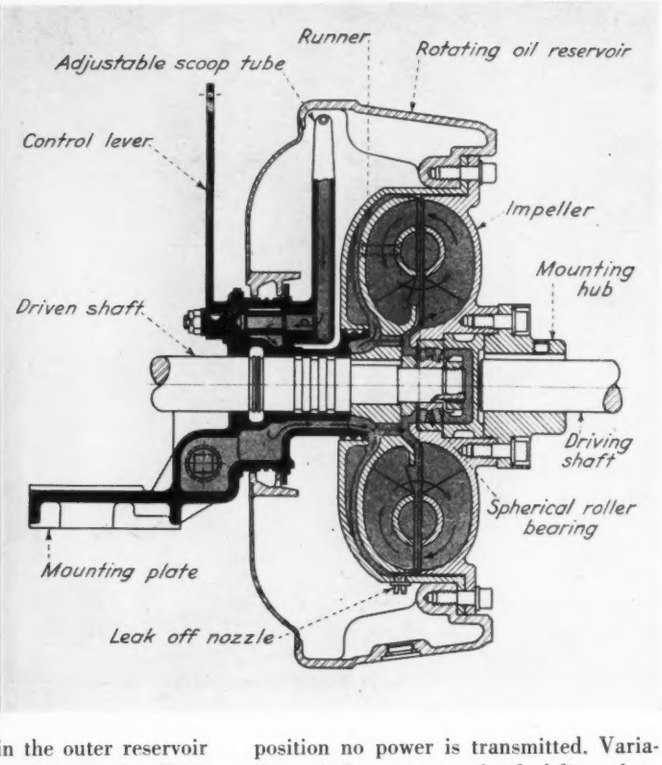
One-piece vanes pass through accurately machined slots in a rotor mounted off-center within a special-shaped bore in the new line of Romec vane-type, rotary, positive-displacement pumps. No springs or pins are required to maintain a smooth-running fit with the pump liner since vanes have a groove machined in their ends into which packing strips are placed to form an effective seal with liner and to automatically take up wear. The bypass and relief valve maintains closely regulated pressure to the carburetor under variable operating conditions. Provision has been made to compensate for super-charger installations in which fuel pressure must be increased exactly in accordance with the increase in air pressure to the carburetor. Valve is a sealed piston type. The new line of Romec aircraft pumps consists of three series: 150, 350 g.p.h., and 640 g.p.h. Performance curves shown below are characteristic for stock units.





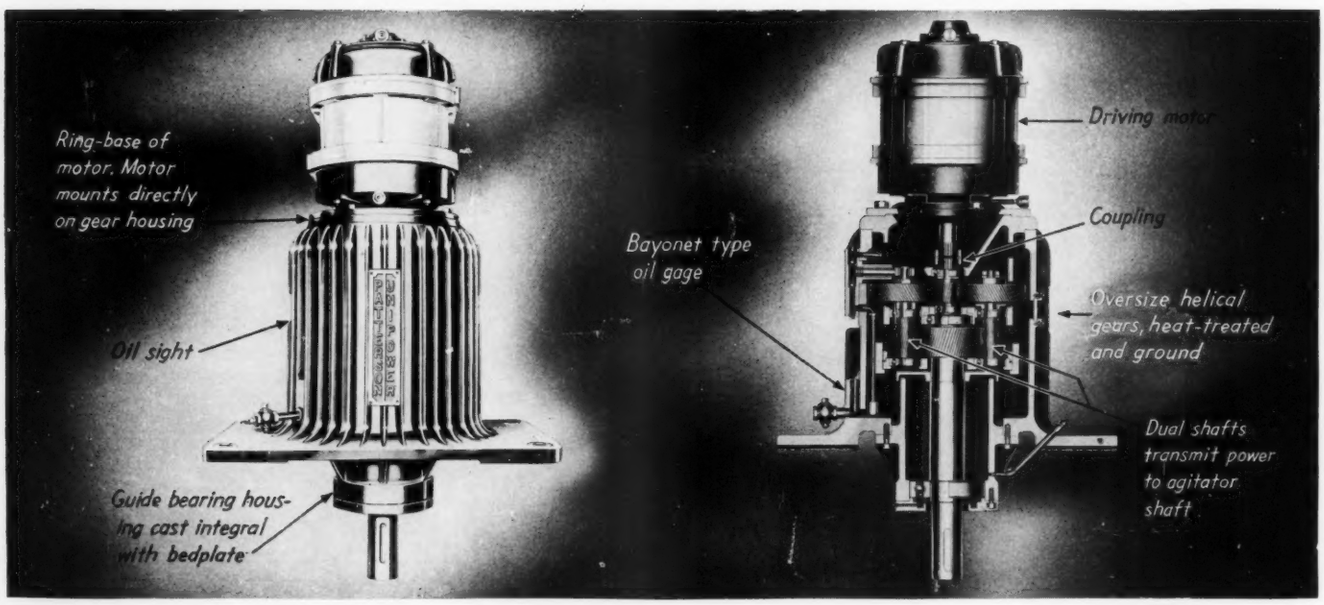
MODERN DESIGNS – Variable-Speed Hydraulic Coupling





Variable speed output is obtained in this American Blower coupling by use of a pivoted scoop tube mounted eccentrically on the stationary manifold. With the coupling rotating, centrifugal force holds the oil in the outer reservoir in an annular ring against the rim. With the scoop tube in its outermost position practically all oil is delivered into the working circuit; at fully retracted position no power is transmitted. Variation of the amount of oil delivered to working circuit varies output. For further data on fluid couplings see P.E. *pages* 96, 466, 524, 1939.

Streamlined Housing for Compact Agitator Drive

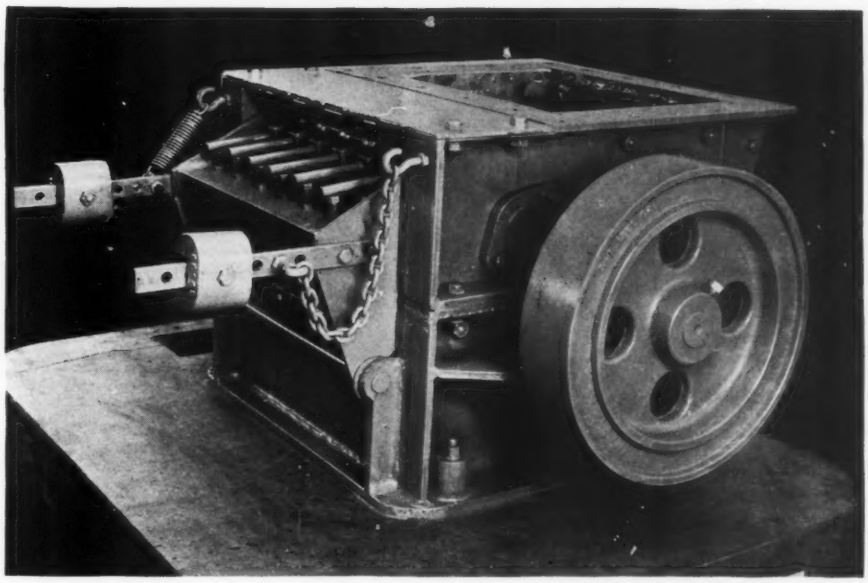

Ring-base of motor. Motor mounts directly on gear housing
Oil sight
Guide bearing housing cast integral with bedplate
Bayonet type oil gage
Driving motor
Coupling
Oversize helical gears, heat-treated and ground
Dual shafts transmit power to agitator shaft

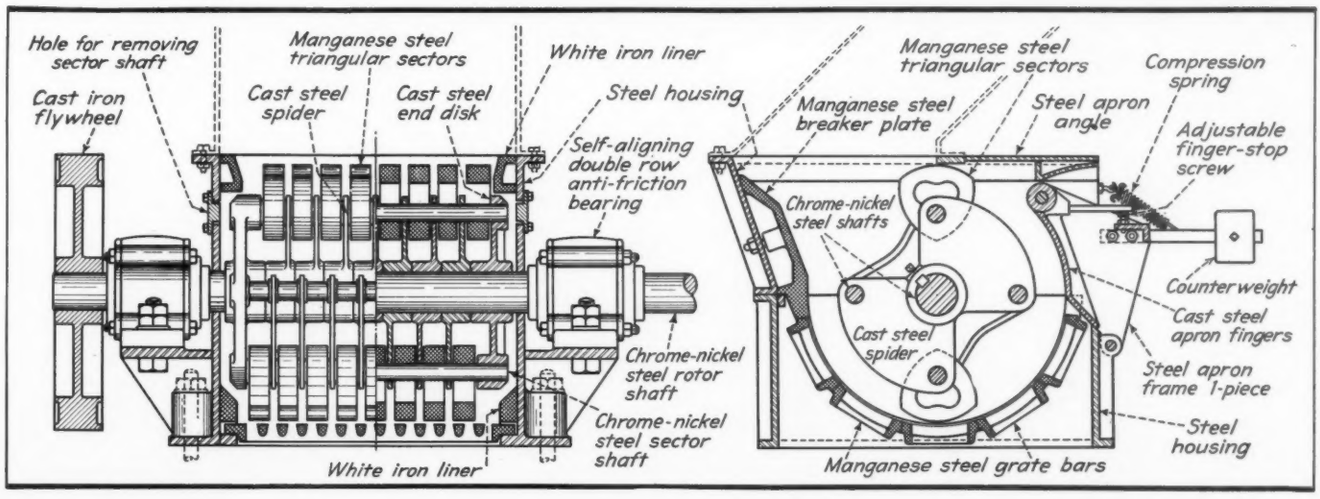
Redesign of housing not only improved appearance in this Patterson agitator drive but also permitted use of an independent oil reservoir and change from immersion to spray lubrication, thus eliminating considerable fluid friction and decreasing the possibility of oil leakage around the output shaft. Increased thrust capacity for the guide bearing was obtained by redesigning housing and using a heavy-duty ball bearing instead of an oilless type.

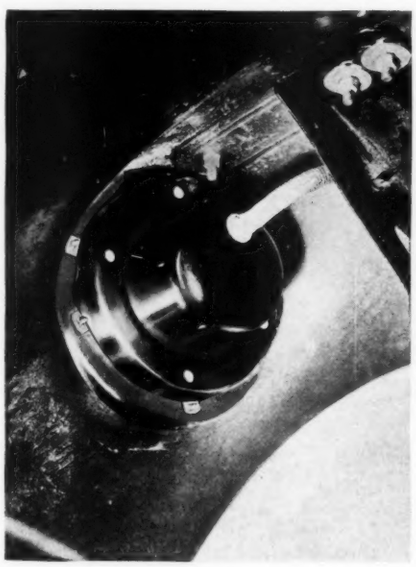
Manganese Steel in Heavy-Duty Crusher Parts

Two balanced gangs of triangular sectors act as crushers in the Stephens-Adamson Knittel ring-type crusher. The sectors are held in position by centrifugal force and are free to recoil under impact with foreign material. Centrifugal action presents the heaviest and least-worn face for crushing duty and equalizes wear on sectors. Balanced rotor and heavy flywheel decrease vibration. Sectors, breaker plates and sizing grate are manganese steel castings; shafts are chrome-nickel steel supported in SKF double-row self-aligning bearings fitted with dust seals.

A series of steel apron fingers, each operating independently, ejects foreign metal fragments and uncrushable material without stopping crusher.

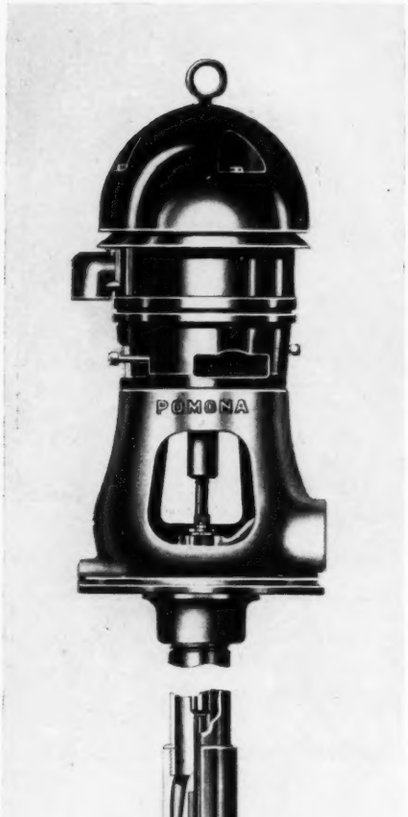




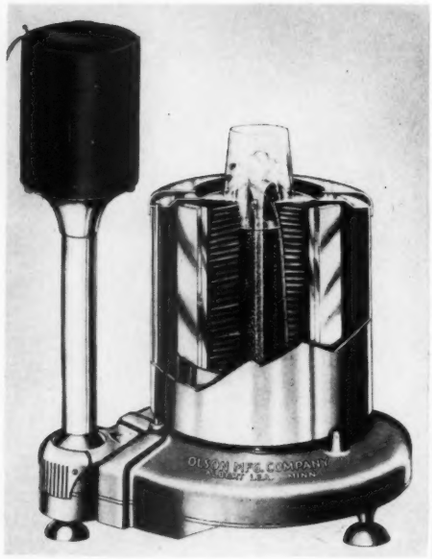


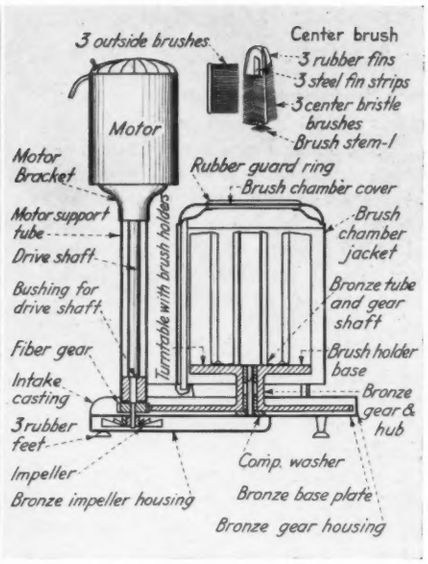
Assembly of sealed-beam headlamp to fender in General Motors and Chrysler cars is by U-shaped Speed Nut fasteners. Nuts require no holding or positioning, can be snapped in place before painting, and do not loosen from vibration. [left]

Advanced constructional features in the Skilsaw Zephyrplane sander include die-castings, forgings, molded parts, ball bearings, oilless bearings, silent chain drive. Housing is aluminum alloy, die-cast. Die-cast idler runs in Oilite bearings mounted in a forged steel yoke. Driver pulley is molded with soft rubber outside layer and hard rubber core. Power is transmitted from the universal motor through worm gears and a 3/16-in. Link-Belt silent chain. [right]

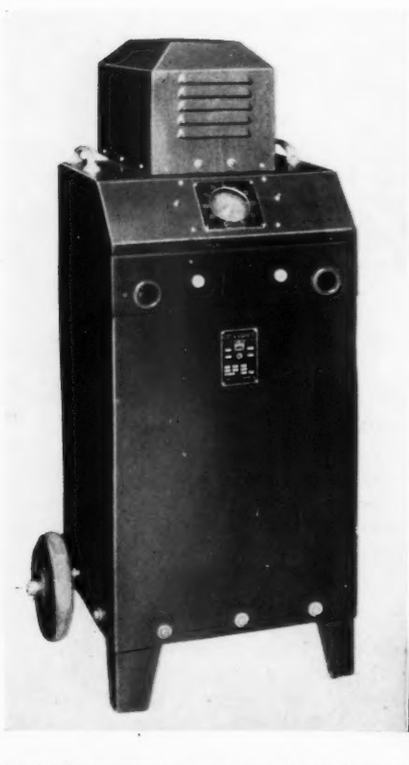


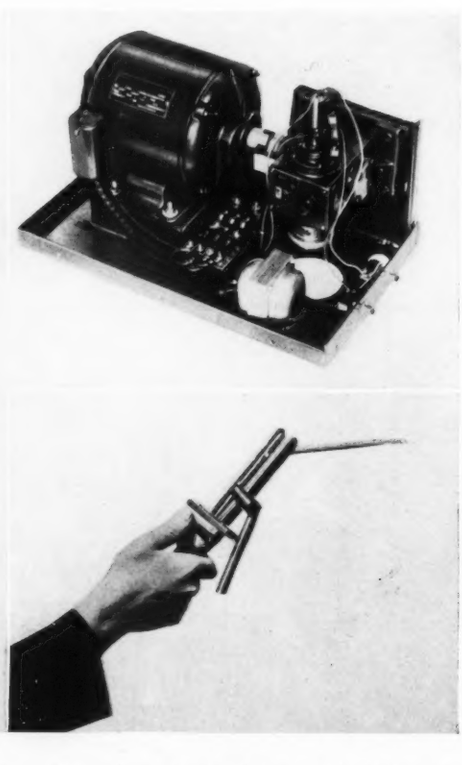
Alloy materials are used extensively in Pomona Little Chief pumps. Motor head and pump bowls are cast from an alloy iron, column pipe is copper-bearing steel, drive shaft is high-strength steel fitted with stainless steel bearing sleeves, strainer is galvanized cast iron. Bearing retainer and impellers are bronze. Water-lubricated rubber bearings in stainless steel bearing sleeves support shaft at 5 ft. intervals.





Portable electric glass-washer unit, designed by Olson Manufacturing Company, is powered by a 1/10 hp., split-phase, Bodine ball-bearing motor which drives brushes through laminated, water-lubricated gears. Parts which are subject to corrosion are stainless steel; other parts are nickel-plated bronze.





Switch mounted on welding tongs controls low-voltage relay circuit through auxiliary cable in the Sterling G-R remote control system. Relays mounted in a ventilated closure on top of the welding set control a reversible motor which drives the current-regulating mechanism through a standard gear reduction unit. An additional switch is provided at the welding set. The remote control system permits current regulation without breaking the arc and gives the welder constant control of welding regardless of the distance between the set and operator.

DRUM AND SHEAVE DESIGN

Effect of Diameter Ratio Factors on Life of Wire Ropes

F. L. SPANGLER

PROBLEMS involving the design of sheaves and drums fall into two classes: those in which dimensions are severely limited by the nature of the machine; and those in which some latitude is allowed in sheave and drum diameters.

In the former class fall such devices as drums on pavers and cranes, tackle blocks, and the closing sheaves of clam-shell buckets. The latter class includes drums and headframe sheaves for mine shaft and skip hoists, elevator drums and sheaves, and other sheave and drum applications where clearances are not highly restricted.

When a wire rope operates over a sheave or drum, bending stresses are introduced into the individual wires of the rope because of the flexing that occurs. As an element of the rope passes onto the sheave, the rope at this point changes from a straight line to an arc of a circle, and the bending stresses caused by this change depend on the flexibility of the rope and the diameter of the sheave which determines the amount of bending. When the same element of rope passes off the sheave, it changes in shape from an arc to a straight line, and in doing so it relieves the bending stresses. These bending stresses are added to the stresses caused by the load, and hence their effect is to increase the total stress in the rope wires at each point of bend. Hence, the total stress in each element of the rope changes as the element passes onto a sheave and again as it passes off.

This frequent change in stress in working ropes causes an embrittlement of the metal which is known as fatigue. Fatigue makes its appearance in the form of minute cracks which soon spread across the wire section, causing a break in the wire. Fatigue is hastened by large changes in stress and by the frequency with which those changes occur. The material of which the wire is made also has a bearing on fatigue, since some metals fatigue more rapidly than others. Also preformed rope is more resistant to fatigue than non-preformed, and therefore where fatigue is an important factor is often specified.

Deterioration of ropes by drums and sheaves is almost entirely caused by fatigue, the factor of wear being relatively unimportant if the sheaves and drums are properly designed and placed in proper relation with one another.

Since the wire rope consists of several strands twisted around a core, and each strand consists of several wires twisted together, it presents a complex mechanism that does not allow of the simple stress calculations that can be applied to solid constructions such as I-beams and rods.

Wire ropes having many small wires are more flexible than those having a few large wires, and the maximum stresses introduced by a given flexure are smaller. Also, when the individual wires and strands of a rope are formed precisely to their final shape in the manufacture of the rope, and therefore lie dormant, as in preformed rope, the individual wires do not press outward on one another, and friction between wires is lessened, with the result that such rope is more flexible and the maximum bending stresses are less.

To guard against fatigue in ropes, the bending stresses must be kept down. Therefore, with sheaves of small diameter, highly flexible rope should be used. If ropes consisting of a few large wires are used, such as 6x7 construction (6 strands of 7 wires each), sheaves and drums of relatively large diameters must be resorted to. The maximum bending stress is proportional to the rope diameter and approximately inversely proportional to the sheave or drum diameter. Therefore, for any given construction of rope, the greater the value of D/d (tread diameter of sheave divided by diameter of rope), the smaller will be the bending stress and the slower will be the development of fatigue.

Table I—Average Sheave and Drum Diameters for Operating Ropes

For 6x 7 rope, 72 times rope diam.
For 6x19 rope, 45 times rope diam.
For 6x37 rope, 27 times rope diam.
For 8x19 rope, 31 times rope diam.

For standing ropes, the above diameters can be almost cut in half.

Table II—Minimum Sheave and Drum Diameters for Operating Ropes

For 6x 7 rope, 42 times rope diam.
For 6x19 rope, 30 times rope diam.
For 6x37 rope, 18 times rope diam.
For 8x19 rope, 21 times rope diam.

Consider a problem involving the moving of a certain load by means of a single rope. The resistance to the pull or lift created by the dead weight of the load and rope, together with the load represented by the acceleration and friction, produce in the rope a certain amount of tension. By multiplying this tension by the factor of safety (never less than 5 for working ropes), the minimum allowable breaking strength of the rope is obtained. By referring to tables published by wire rope manufacturers which give the breaking strength of ropes, the size of rope can be determined.

Knowing the diameter of the rope, and its construction and material, calculations can be made to determine the most desirable diameter of the drum and sheaves. Before doing so, however, it is well to point out the controversial nature of this subject. Although wire-rope manufacturers are generally in agreement on recommended sheave and drum diameters, yet some manufacturers of rope-using equipment claim that these recommended diameters would be impracticable because of space limitations and considerations of weight.

The respective merits of both sides of this controversy will not be gone into here, other than to point out that rope manufacturers, though adhering to their original position, point out that smaller drums or sheaves may at times be justified even at the expense of lessened rope life; and the manufacturers of rope-using equipment now recognize that until the "perfect" rope is produced, combining the qualities of strength and flexibility with resistance to fatigue and abrasion, the only practical solution to the problem of rope deterioration are sheaves and drums that do not introduce high bending stresses in the rope.

Drum and sheave design problems may seem simple to the uninitiated, but at best they are complicated by various factors. The determination of sheave and drum diameter is, in the final analysis, a matter of balancing rope savings against other economies. A change from small to large diameters for headframe sheaves will save more dollars in rope than a similar change in an appli-

cation such as equalizing sheaves on cranes, because the headframe sheave introduces large bending stresses of an alternating character into the rope throughout most of its length, and hence fatigue breaks soon occur in the wires with consequent discarding of the entire rope, while in the case of the equalizing sheave on a shop crane the movement of the rope over the sheave is very slight and hence the fatiguing action is slow and rope life is comparatively long. Therefore, the cost of large sheaves may be justified in the headframe, but not in the crane equalizing sheave.

Another problem in economics, but of a different nature, is the case of sheaves over which is operated only a small section of rope located near the end, such as the sheaves on the closing mechanism of a clamshell bucket. Though these sheaves are necessarily of small diameter because of space limitations, yet if large sheaves were practicable they would scarcely be justified because the practice of cutting off the damaged end of the rope at frequent intervals, and compensating by letting out more rope at the drum end, results in only a small waste compared to discarding the entire rope.

Though mention has been made of the fatiguing action to wire rope caused by bending stresses when passing over sheaves and drums, one must not overlook the fact that fatigue in ropes is also frequently caused by vibration and whipping. Fatigue breaks in the individual wires of ropes are especially apt to occur near the shackled ends, and is accellerated by improper shackling. It is poor economy to try to eliminate all fatigue at the sheaves if at the same time severe fatiguing action is being set up elsewhere.

The impossibility of setting up universal standards for sheave and drum diameters becomes at once apparent. Representative averages based on drum and sheave sizes being used on elevators, dredges, derricks, cranes, mine hoists, and certain other equipment are given in Table I, while Table II gives diameters below which rope operation will generally be found unsatisfactory because of the greatly shortened life of the rope. Since the figures given in Tables I and II are based on average data, they should not be used uncompromisingly in design, but should serve only as a guide.

The sliding of the strands of a wire rope over each other accounts for the flexibility of the rope and for its ability to handle heavy loads in severe service. There is a certain minimum bending radius below which the strands of a rope clamp each other in a vise-like action, causing the strands to lose their independence of motion. This diameter, known as the critical diameter, marks the sheave size below which the rope strands cannot slide past each other to adjust themselves to the bend, and consequently constructional restrictions are set up in the rope, resulting in excessive internal stresses. Ropes operating over sheaves of such small diameters go to pieces quickly. Table III give these critical diameters.

Table III—Critical Diameters for Sheaves and Drums

Rope Construction	Critical Diameter
6x7	28 × rope diam.
6x19 Seale	20 × rope diam.
6x19 Warrington	16 × rope diam.
6x19 Filler Wire	16 × rope diam.
6x37	14 × rope diam.
8x19 Warrington	14 × rope diam.
8x19 Seale	16 × rope diam.

Now consider the situation where the sheaves and drums are so limited in size by unavoidable restrictions that the only possibility to reduce rope fatigue lies in reducing the size of the rope or changing its construction. Rope size can be reduced by using sheaves or tackle blocks at some fixed point and at the load, and reeving the rope through these sheaves or blocks in several parts. This greatly decreases the lead line pull, not exactly in proportion to the number of rope parts, but in accordance with the factors given in Table IV, which also gives efficiency of tackle equipment.

While the smaller tension in the lead line allows a reduction in the size of the rope, one must not overlook the fact that the length of rope wound onto the drum during a complete lifting cycle is proportional to the number of rope parts, and the speed of the rope is equal to the speed at which the load is handled, multiplied by the number of rope parts. Therefore, the drum capacity must be increased to take care of the increased length of rope; and, to maintain the same rate of load travel, the peripheral speed of the drum increased in like proportion.

Another solution to the problem is to omit the tackle blocks and to employ two or more ropes in parallel, each attached to the load independently of the other ropes, or through equalizing bars or equalizing sheaves. The rope speed, then, is the same as with a single rope, but the rope diameter is decreased and the drum capacity increased as in the example where tackle blocks were considered.

The material used for sheave and drum treads has an important bearing on the service obtained from the rope as well as from the sheaves and drums. Where

P = unit pressure of the rope on the drum or sheave, in lb. per sq.in.
T = rope tension, in lb.
d and D = diameters, in in., of rope and drum, respectively

the maximum unit radial pressures of the rope on the drum or sheave can be determined from $P = 2T/dD$.

By comparing the value of P, as obtained above, with the safe unit bearing pressures on various materials in sheaves and drums as given in Table V, a material can be selected that will safely stand the unit bearing working pressure.

Where sheaves or multiple U-grooved wheels or drums are employed to develop a tractive force, as in traction elevators and endless drives, consideration must be given to the number of laps, or half laps, required to develop the necessary traction without slippage.

If T_1 is the tension in the tight side of the rope, in lb., and T_2 is the tension in the loose side, in lb., then the ratio T_1/T_2 is expressed as $T_1/T_2 = \varepsilon^{\mu\theta}$ where ε = base of the Napierian system of logarithms = 2.718282+; μ = coefficient of friction of rope in groove (see Table VI); and θ = arc of contact of the rope with the sheave, expressed in radians.

It is evident from the foregoing formula that the diameter of the sheaves or wheels has no effect on the tractive power of the rope, unless it affects the arc of contact, which is seldom the condition.

To facilitate calculations by the foregoing formula, in cases where θ is equivalent to 180 deg. or a half lap, the values in Table VII will be found useful.

Where a greater tractive force than that given by a wire rope operating in a U-groove is desired, the V-groove may

Table IV—Lead Line Pull Factors and Efficiencies for Hoist or Fall Ropes

Number of parts of rope	2	3	4	5	6	7	8	9	10
Efficiency, %	96.15	92.45	88.89	86.48	82.19	79.03	75.99	73.06	70.25
Lead line pull factor, %	52.0	36.05	28.12	23.39	20.27	18.07	16.45	15.21	14.23

The stress in the lead line equals the load multiplied by the lead line pull factor.

Table V—Safe Unit Bearing Pressures of Ropes on Various Materials in Sheaves
Pounds per square inch

	Regular Lay Rope				Lang Lay Rope			
	6x7	6x19	6x37	8x19	6x7	6x19	6x37	Remarks
Wood	150	250	300	350	165	275	330	On end of grain of beech, hickory, gum
Cast Iron	300	500	600	700	350	550	660	Average Brinell hardness of 160
Carbon Steel Casting	550	900	1,075	1,250	600	1,000	1,180	0.30–0.40 carbon: average Brinell hardness of 160
Chilled Cast Iron	650	1,100	1,325	1,550	715	1,210	1,450	Not advised unless surface is uniform in hardness
Manganese Steel	1,500	2,500	3,000	3,500	1,650	2,750	3,300	Grooves must be ground and sheaves balanced for high-speed service

be used, having sloping sides that form an included angle between 30 and 45 deg. If this included angle be designated by ϕ, then the relation of T_1 to T_2 becomes $T_1/T_2 = \epsilon^k$ where

$$k = \frac{\mu\,\theta}{\sin \frac{1}{2}\phi}$$

A rope operating in a V-groove should be of a softer material than that of which the groove is made; otherwise the rope will wear a seat in the groove, thereby reducing the pinching action of the groove and nullifying its effect.

With free-running sheaves, the arc of contact between the rope and the sheave should not be so small that the friction developed is not sufficient to overcome the inertia of the sheave, with the result that acceleration and deceleration may cause a slipping of the rope along the sheave groove, thus causing wear of both the rope and sheave. Friction between rope and sheave can be increased by shifting the position of the sheave so as to increase the angle of contact of the rope with the sheave. Otherwise, it will be necessary to resort to a smaller sheave or one made of a lighter material.

Table VI—Coefficient of Friction μ for Ropes Operating on U-Grooves of Various Materials

Condition of Rope	Iron or Steel	Wood	Rubber or Leather Lining
Dry	0.170	0.235	0.495
Wet	0.085	0.170	0.400
Greasy	0.070	0.140	0.205

Table VII—Values of $\epsilon^{\mu\theta}$ for Multiples of 180 deg.

	Number of Half Laps About Wheels					
Value of μ	1	2	3	4	5	6
0.070	1.246	1.552	1.934	2.410	3.003	3.741
0.085	1.306	1.706	2.228	2.910	3.801	4.964
0.140	1.552	2.410	3.741	5.808	9.017	13.998
0.170	1.706	2.910	4.964	8.467	14.445	24.641
0.235	2.092	4.378	9.160	19.166	40.100	83.902

Table VIII—Proper Groove Diameters

	Tolerance of Groove Diameters	
Nominal Diameter of Rope	Minimum	Maximum
½ in. and smaller	+1/32 in.	+3/32 in.
9/16 in. to 1 in.	+1/16 in.	+⅛ in.
1 1/16 in. to 2 in.	+3/32 in.	+3/16 in.
Over 2 in.	+⅛ in.	+¼ in.

Wherever possible, sheaves and drums should be so positioned as to avoid reverse bends, since a reverse bending action is very favorable to the development of fatigue and is therefore highly destructive to ropes. If a reverse bend cannot be avoided, the sheaves should be a considerable distance apart, allowing a long length of rope between adjacent bends, and the sheave diameters should be increased if possible. Such measures will counteract much of the destructive action of reverse bending.

Sheaves should be mounted in perfect alignment; otherwise, excessive wear of both rope and sheave will occur. The lead from a sheave to a winding drum should be as long as practicable in order to keep the fleet angle small, thereby preventing rubbing of the rope on the sides of the sheave groove. A large fleet angle also often results in uneven winding and piling of the rope on the drum. Where space permits, a lead of 40 ft. per foot of drum fleet is desirable.

The design of the grooves in drums and sheaves is important. The size and contour of the groove may affect greatly the life of the rope. Good design demands that the grooves be slightly oversize compared with the diameter of the rope, and yet not so large as materially to reduce the bearing surface of the rope on the groove throat. In grooving a drum or sheave, one should keep in mind that the diameter of a rope is slightly larger when new than its nominal diameter. If groove diameters are machined equal only to the nominal diameter of the rope, the rope will be pinched by the sides of the groove until it has had time to seat itself by wearing the groove. Pinching caused by small grooves is highly destructive to rope.

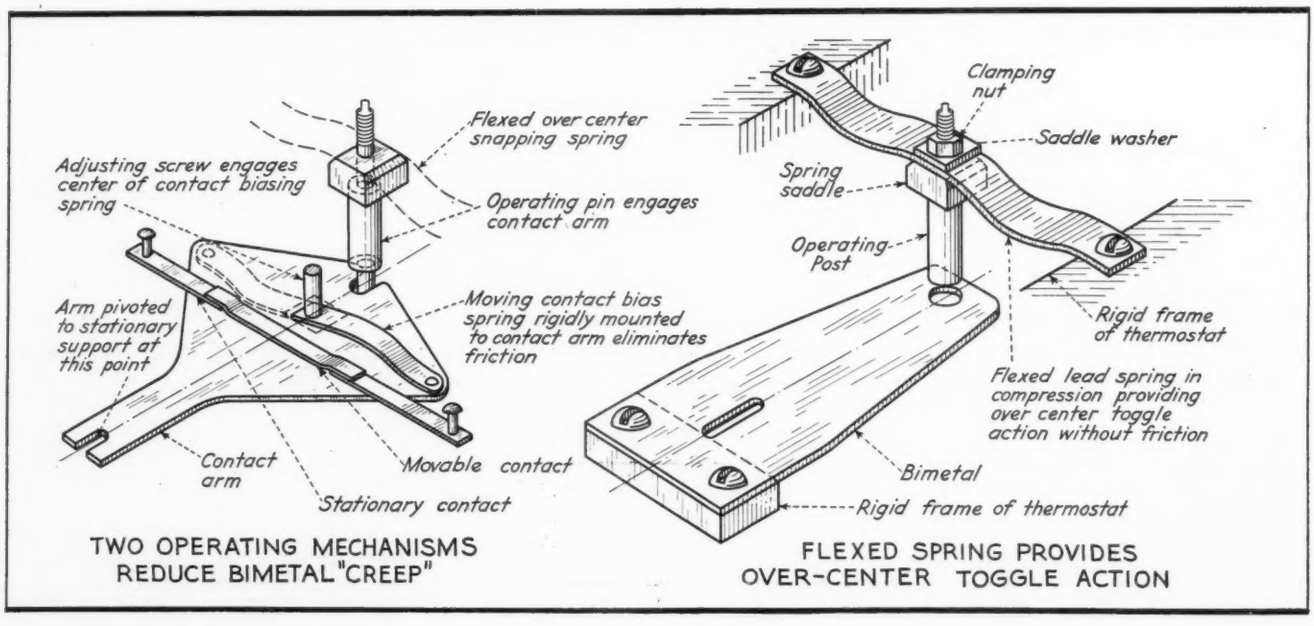
In Table VIII is given the tolerance of groove diameters for the complete range of rope sizes. Grooves made according to these specifications will allow ropes to work freely, even though the ropes be slightly oversize in accordance with manufacturing tolerances.

THERMOSTAT DESIGN

Improved Mechanisms Increase Operating Efficiency

E. K. CLARK
Westinghouse Electric & Manufacturing Company

IMPORTANT factors to be considered in the design of small bimetal electrical mechanisms include contact action, reduction of friction, sensitivity and ease of adjustment. Electrical considerations such as choice of proper contact material and size of contacts to be used will not be discussed in this article. The mechanical problems in obtaining efficient contact action involve the use of reasonably high contact pressures and the design of a mechanism to store up the energy required for snap-action. This is generally accomplished with some sort of spring; if spring pressure is insufficient, arcing and pitting of contacts may result and positive action may not be obtained. Sensitivity can be increased and operation errors minimized by reduction of the number of pivots and fulcrums with their attendant friction. In the following examples, the four mechanical improvements illustrated schematically show methods used to obtain more positive action, reduce friction, increase sensitivity and to provide for easier adjustment.

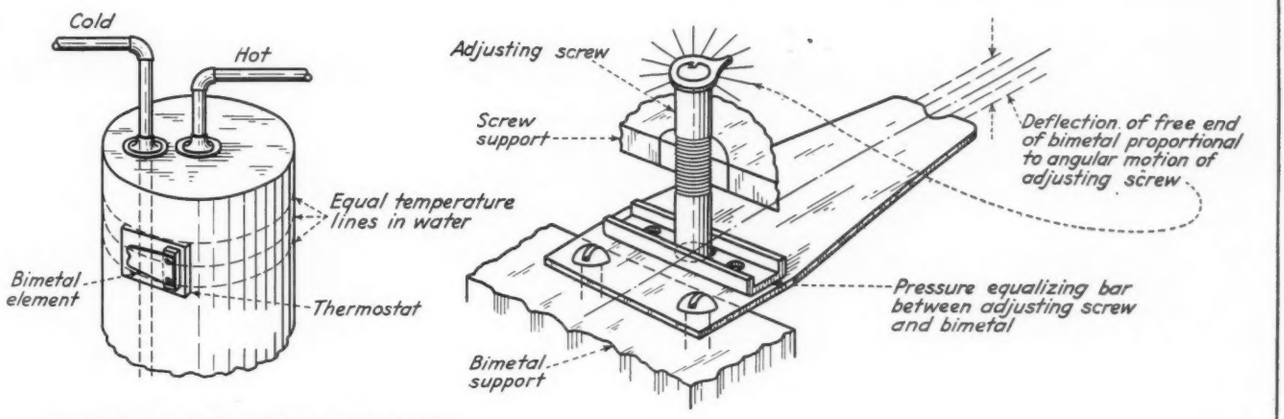

Flexed over center snapping spring
Adjusting screw engages center of contact biasing spring
Operating pin engages contact arm
Arm pivoted to stationary support at this point
Moving contact bias spring rigidly mounted to contact arm eliminates friction
Contact arm
Movable contact
Stationary contact
TWO OPERATING MECHANISMS REDUCE BIMETAL "CREEP"
Clamping nut
Saddle washer
Spring saddle
Operating Post
Rigid frame of thermostat
Flexed lead spring in compression providing over center toggle action without friction
Bimetal
Rigid frame of thermostat
FLEXED SPRING PROVIDES OVER-CENTER TOGGLE ACTION

Two separate but interdependent mechanisms comprising a thermal operating unit and a contact operating mechanism. Thermal operating unit consists of the bimetal and cooperating over-center biasing spring assembly, which operates interdependently of the contact operating mechanism, and permits the building up and storing of energy prior to contact operation. This establishes positive switching action and prevents any creep of the thermal operating mechanism from affecting the contact operating mechanism.

Contact operating mechanism consists of a contact arm and a separate and independent over-center spring. Contact arm is supported and independently biased by its over-center spring equally into its open and closed positions, thus maintaining separate and independent minimum pressure at point of thermostat operation. This construction therefore does not permit contact pressure at the snapping point of the thermostat to approach zero, but definitely maintains a minimum value which will assure positive action.

Electrical switching is accomplished by the thermal operating mechanism impinging on the contact operating mechanism after the former has stored an appreciable amount of energy. Important advantages are that a low differential thermostat will positively operate heavy duty contacts without permitting these contacts to pass through zero contact pressure and will also compensate for an occasional thermostat which may be improperly adjusted and which may creep a short distance before it snaps. This creeping action of an improperly adjusted thermostat will not be transmitted to the contacts in the structure.

New principle of over-center motion, based on flexing within the spring itself, is being applied to thermostats to improve sensitivity by eliminating the usual pivots and fulcrums with attendant friction errors. The over-center spring is rigidly mounted on the supporting frame, preventing this key spring member shifting in its adjustment or jarring out of position through rough handling or in shipment. A further fundamental improvement which has accompanied the use of the integral over-center spring blade, and was made possible by it, is the precise control of the over-center spring tensions this blade applies to the bimetal. These tension components so exactly match the physical properties of the bimetal that quantity production of a positive snap-acting thermostat without incorporating a differential adjustment is now made possible.

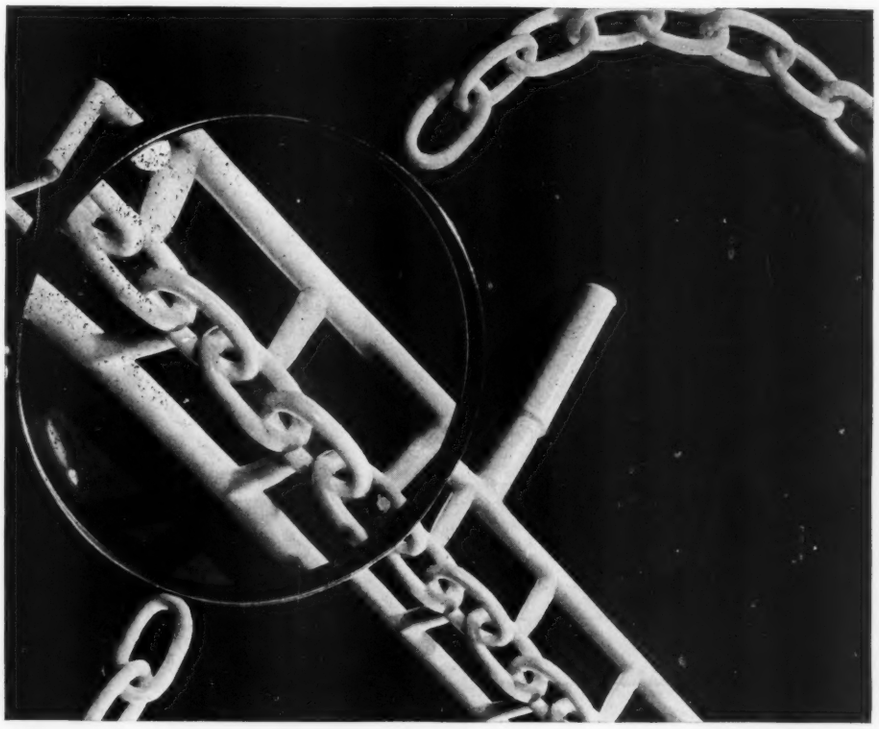


METHOD OF MOUNTING INCREASES THERMOSTAT SENSITIVITY

SCALE CALIBRATED TO WATER TEMPERATURE

Bimetal with large surface area in close radiant heat exchange with the hot water tank surface to which it responds has greatly increased thermal sensitivity. In the most recent development, the water heater thermostat has a bimetallic member mounted parallel and closely adjacent to the bottom, or mounting surface of the thermostat, and has exposed to the heated surface of the tank a surface area of approximately two to three times that ordinarily obtained. Because heat can be conducted to and from the bimetal only through the comparatively small area where the fixed end of the bimetal is attached to the supporting frame, heat transmitted between the bimetal and the tank surface by radiation is used to obtain maximum sensitivity. This relationship is further improved by mounting the bimetal with its long axis across the tank or parallel to the constant temperature lines of the water as it stratifies vertically in heating and cooling. Thus the bimetal responds through its entire length to a uniform temperature of water rather than to a temperature gradient.

Scale on water heater thermostat is calibrated to indicate water temperatures. An adjusting screw controls the operating temperature of the bimetal. A true scale thermostat with precisely controlled, true temperature relationship between bimetal and water temperature must maintain a straight line relationship of bimetal temperature to angular movement of adjusting screw over full scale. This relationship is obtained by a mechanical system that makes normal deflection of free end of bimetal proportional to angular movement of screw.



One-Piece Plastic Chain Injection Molded

By ingenious sectional mold design, these Tenite chains are the first one-piece plastic chains to be produced by injection molding. Chains, shown in both cast and finished form, are molded by Tilton & Cook Company, using a four-piece mold that eliminates cutting, threading, and cementing of links, previously necessary in making plastic molded chains. Severing of links from runners is the only finishing required. Chains are strong and light in weight, and come from the mold with a high polish. Chains are available in a full range of transparent and translucent colors.

CANTILEVER BEAMS

How to Design for a Given Deflection at Any Point

C. C. STREET
Federal Products Corporation

A DIFFICULTY often met with when calibrating and testing instruments of an indicating nature is in obtaining a measured motion of short range and readable to fine units. Such instruments as dial indicators, the mechanism of recorders, relays, contacts and snap switches of the "Micro Switch" type are a few of the devices that call for this kind of check. In the conventional reduction lever, at ratios of more than 10 to 1 backlash and pivot pressure produce uncertainties. If a micrometer is used with a reduction gear to turn the barrel, backlash in the gearing, even if spring loaded because friction changes tooth pressure, and the creepage of oil in the thread will often cause errors greatly in excess of the least division to which it is desired to read.

A mechanism that has been used with commendable success to obtain these small increments is based on the deflection of a simple cantilever beam. If such a beam consists of a strip of spring steel firmly supported at one end and the other deflected a known amount, the deflections at various points in terms of the end deflection will be in ratios ranging from 1:1 to ∞:1 as we go from the free end to the fixed end of the beam.

If y_m is the known deflection, then at any distance x from the origin O the deflection y_x holds a definite ratio to y_m. To evaluate this ratio, let us use the formula for the simple cantilever beam.

The deflection curve of such a beam as shown in Fig. 1 is expressed by:

$$y_x = -\frac{W}{6EI}(x^3 - 3l^2x + 2l^3) \qquad (1)$$

Where W = load in lb. acting along Y axis in a negative direction

E = modulus of elasticity of the material

I = moment of inertia of beam

x = distance from Y axis to point where deflection y_x is measured

l = length of beam

In this particular application we are interested in the ratio that exists between y at $x = 0$ and y at some value of x that lies between O and point C. The problem is to solve for a value of x that will give the desired ratio between y at $x = 0$ and y_x at some other point.

The expression for y_{max} when $x = 0$ is:

$$y_m = -\frac{Wl^3}{3EI} \qquad (2)$$

Since we are interested in the ratio of $\frac{y_x}{y_m}$ Equation 1 can be divided by Equation 2.

$$\frac{y_x}{y_m} = \frac{-\frac{W}{6EI}}{-\frac{Wl^3}{3EI}}(x^3 - 3l^2x + 2l^3)$$

$$\frac{y_x}{y_m} = \frac{1}{2l^3}(x^3 - 3l^2x + 2l^3) \qquad (3)$$

Rearranging:

$$x^3 - 3l^2x + 2l^3 - \frac{2l^3\, y_x}{y_m} = 0 \qquad (4)$$

When Equation 4 is solved as a cubic

$$x = -2l\cos\left(\frac{\cos^{-1}(1 - y_x/y_m)}{3} + 240°\right) \qquad (5)$$

Table I gives a solution for some of the common ratios.

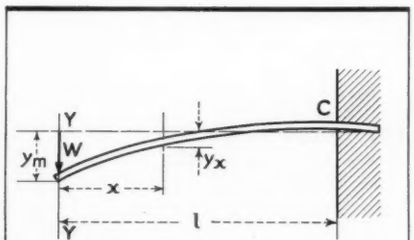


Fig. 1—Deflection curve of a simple cantilever beam loaded at its free end

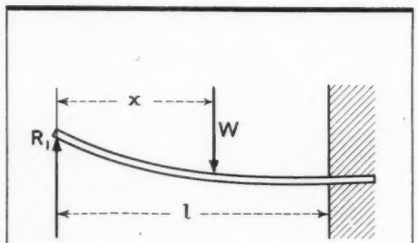


Fig. 2—Cantilever beam with intermediate load W, which is the force exerted by the device under test, and the reaction R_1 of the micrometer spindle

Applying this principle, an instrument as shown in Fig. 3 was developed.

By experimentation and evaluating for the foreshortening of the beam it was found that if the deflection at y_m did not exceed 1/40 of l the relationship held to a degree of accuracy such that a ratio of $y_m/y_x = 100$ was easily usable. In this instrument a micrometer reading in 0.0001 in. was used so that with a ratio of 100 to 1 each division was 10^{-6} in. at x. Care must be exercised so that the steel contact ball is as nearly in the center of the micrometer spindle as possible to eliminate side motion due to friction between the ball and the face of the spindle.

If the device being tested exerts a pressure of measurable magnitude on the beam it will be deformed from its true cantilever shape. This deformation would be analyzed as a beam fixed at one end and supported at the other.

The intermediate load W is the force exerted by the device under test, and R_1 is the reaction on the micrometer spindle. After equilibrium has been established the beam will act as a cantilever as previously described since the principle of super-position holds.

In many instruments the force exerted varies with the deflection. The effect of this is unavoidable but can be reduced to a minimum by using as heavy a beam as the necessary deflection will allow.

Referring to Fig. 2 the deflection Δ at the point of application of the load W is given by

$$x = \frac{W_x x^2 (l - x)^3}{E\, l^2\, b\, d^3}(3l + x) \qquad (6)$$

Where W_x = force of instrument on beam lb.

x = distance from point of application to free end in inches

l = length of beam inches

E = modulus of elasticity

b = width in inches

d = thickness in inches

This deflection Δ for a given force, reaches a maximum when $x = 0.414l$. It is quantitatively expressed by:

$$\Delta_{max} = 0.11764\,\frac{W_x l^3}{E\, b d^3} \qquad (7)$$

It can be seen from this that the deflection increases as the cube of the

length and decreases as the cube of the thickness.

The limit of these values is determined by the action of the beam as a cantilever and the necessary deflection, y_m. Here the stress is given by:

$$f = \frac{3\ y_m\ E\ d}{2l^2} \qquad (8)$$

Where f = stress in lb. per sq.in. A good working stress for high quality spring steel is 50,000 lb. per sq.in., which will keep hysteresis to a minimum.

There is still another fact to be considered, namely, the deformation and stress produced at the contact between the steel ball and the micrometer spindle.

Assuming the same values for E and Poisson's ratio for both spindle and ball, the compressive stress at their contact is given by

$$f_c = 0.616\sqrt[3]{\frac{W\ E^2}{D^2}} \qquad (9)$$

Where W = force of ball against spindle in lb.
D = diameter of ball in in.
f_c = compressive stress at contact

From the experience of ball bearing manufacturers, 300,000 lb. per sq.in. can be used as a safe stress when there is no rolling motion.

The necessary length of the beam can be determined by $l = 40\ y_m$. If l is assigned, the thickness of the beam can be determined so that the limiting stress or the maximum allowable value of W will not be exceeded. To simplify calculations, rearranging Eq. 9 gives:

$$W = 0.0379\ \frac{D^2\ f_c^3}{E^2} \qquad (10)$$

Substituting in Eq. 2 the value $I = bd^3/12$ and solving; using the above calculated value for W:

$$d = \sqrt[3]{\frac{4\ W\ l^3}{y\ E\ b}} \qquad (11)$$

By solving this equation a thickness is found that will give a safe force on the ball contact.

By rearranging Eq. (8) and solving:

$$d = \frac{2\ f\ l^2}{3\ y\ E} \qquad (12)$$

Substituting in this equation the known values and the maximum allowable stress gives the value of d for the maximum stress. For a given deflection, the greater the value of d the greater will be the stress. The smaller of the two values of d computed by Eq. 11 and Eq. 12 should be used if neither W or f are to exceed the limiting values.

TABLE I—Deflection Ratios

$\frac{y_x}{y_m}$	$\frac{x}{l}$
0.005	0.9422
0.010	0.9172
0.020	0.8822
0.050	0.8113
0.100	0.7293

Example:

$y_m = 0.200 \qquad l = 0.2 \times 40 = 8$

$\frac{y_x}{y_m} = .01$ (selected)

$x/l = 0.9172$ (from table)

$x = 8 \times 0.9172 = 7.337$

Let: $b = 1 \qquad f = 5 \times 10^4$ lb. per sq. in.
$D = 0.25 \qquad E = 3 \times 10^7$

$$W = 0.0379\ \frac{(0.25)^2\ (3 \times 10^5)^3}{(3 \times 10^7)^2} = 1.875 \text{ lb.}$$

$$d_1 = \sqrt[3]{\frac{(4)\ (1.875)\ (8)^3}{(0.2)\ (3 \times 10^7)\ (1)}} = 0.0862 \text{ in.} \quad \text{(From Eq. 11)}$$

$$d_2 = \frac{2\ (5 \times 10^4)\ (6.4 \times 10)}{3\ (2 \times 10^1)\ (3 \times 10^7)} = 0.355 \text{ in.} \quad \text{(From Eq. 12)}$$

So as not to exceed the contact force of 1.875 lb., the 0.0862 in. value of d is used and the stress will be well below the limit of 5×10^4. Its actual value will be:

$$f = \frac{3_y\ E\ d}{2l^2} = \frac{3 \times 0.2 \times 3 \times 10^7 \times 0.0862}{2 \times 64}$$

$= 12{,}100$ lb. per sq. in.

As calculated by Eq. 8.

This method of calculation can be applied to the design of any type of cantilever beam of constant cross-section and offers a simple method for determining the vertical deflection at any point in a cantilever beam in terms of the end deflection.

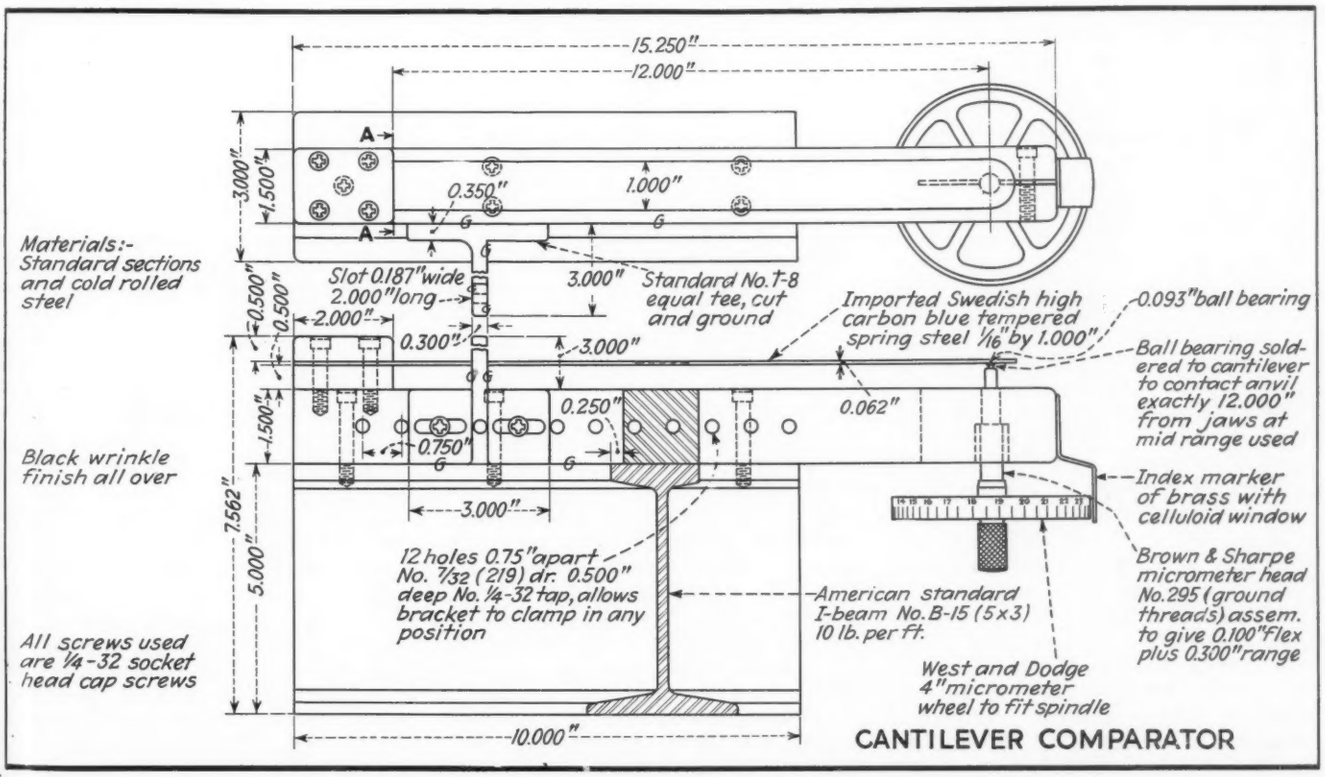

15.250"
12.000"
3.000"
1.500"
A
A
0.350"
1.000"
Materials:- Standard sections and cold rolled steel
Slot 0.187" wide 2.000" long
3.000"
Standard No. T-8 equal tee, cut and ground
Imported Swedish high carbon blue tempered spring steel 1/16" by 1.000"
0.093" ball bearing
0.500"
0.500"
2.000"
0.300"
3.000"
Ball bearing soldered to cantilever to contact anvil exactly 12.000" from jaws at mid range used
0.062"
0.250"
1.500"
0.750"
Black wrinkle finish all over
Index marker of brass with celluloid window
3.000"
7.562"
5.000"
12 holes 0.75" apart No. 7/32 (219) dr. 0.500" deep No. 1/4-32 tap, allows bracket to clamp in any position
American standard I-beam No. B-15 (5x3) 10 lb. per ft.
Brown & Sharpe micrometer head No. 295 (ground threads) assem. to give 0.100" flex plus 0.300" range
All screws used are 1/4-32 socket head cap screws
West and Dodge 4" micrometer wheel to fit spindle
10.000"
CANTILEVER COMPARATOR

Fig. 3—Cantilever comparator designed on the basis of the equations developed in this article

THE ELECTRIC EYE IN INDUSTRY

Applications of Standardized Units

JOHN A. LONG
Photoswitch, Incorporated

THE photoelectric field is often associated with complicated and delicate apparatus, specially tailor-made at considerable expense. In many cases delicately engineered equipment of a laboratory type requires particular conditions of local illumination and constant voltage not available in the average plant. This has in many cases influenced designers not to recommend photoelectric controls, even when conditions are such that the electric eye could do the job better than any other means.

However, electric eye equipment is available today which can be successfully used under ordinary operating conditions found in industry. These controls have been engineered to operate successfully under high temperature, humidity, and vibration conditions.

In the typical uses of the electric eye discussed below, no attempt has been made to make the listing all inclusive, but other applications will readily suggest themselves.

The most desirable design of safety device is that which will interfere the least with press production. Light beam protection is generally more satisfactory than a mechanical arm in that it does not cut off the operator's view, and it may be incorporated as a press control as well as a safety device. If the press is operated by an electric clutch and brake, the output of the electric eye may control these directly. Installation is relatively simple. For presses equipped with mechanical clutches, the electric eye control can operate a lock on the foot pedal and prevent the clutch from being thrown while the operator is in danger.

Standard photoelectric equipment in the form of banks of photo-cells operating from a curtain of light to fit any cross-section can be used to safeguard the working area of machinery.

With certain machinery and industrial processes it is sometimes necessary to keep workmen at a reasonable distance from the operations. With standard photoelectric equipment, a light beam fence up to 100 feet in length may be set up and readily adapted by mirrors to any reasonable length and shape. When a workman crosses the light beam, the control may operate an alarm, or turn off the machine itself. In one instance, photoelectric equipment has been adapted as a safeguard in wire drawing processes. The control light beam is projected 50 ft. behind the entire length of a wire draw bench. In case the operator becomes entangled in the wire while making adjustments, his body will cut off the light beam and immediately stop the machinery.

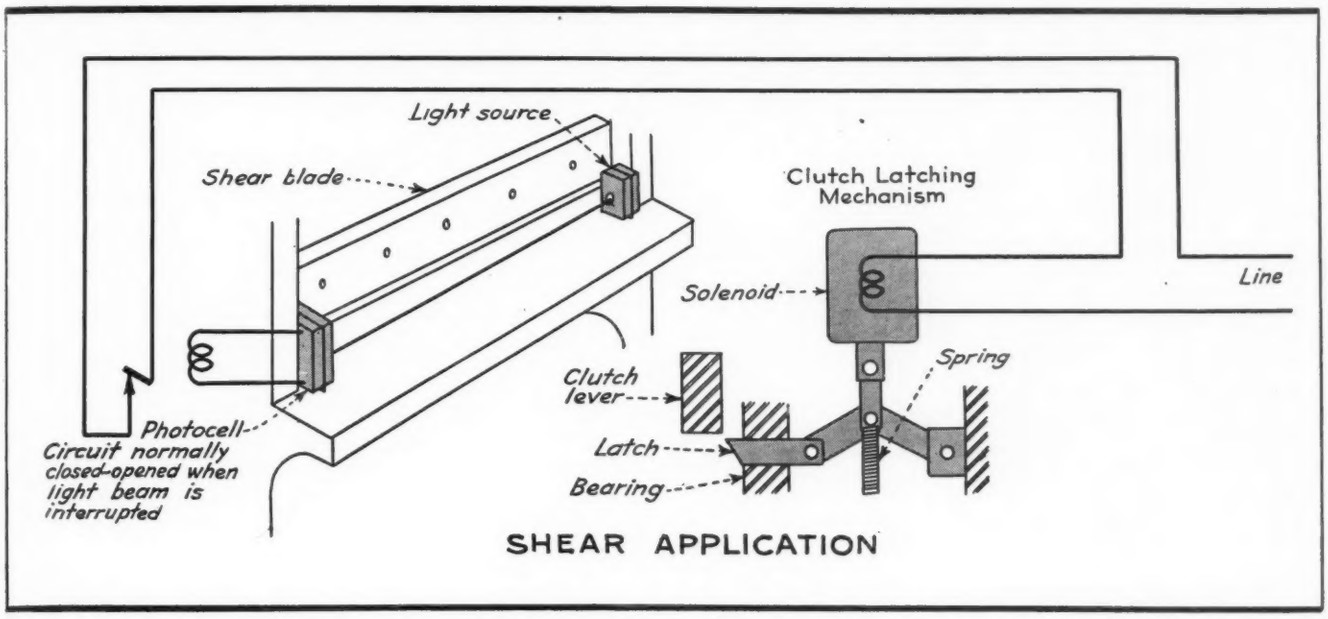
One copper wire manufacturer uses the electric eye to detect kinks and other irregularities in freely unwinding cable on its way to the winding reel.

Photoelectric control also provides speed compensation in the stranding of wire cables. The stranding is performed at a constant speed, and the cable is then rolled about a capstan, also operating at a constant speed, and finally is fed to a large spool. The spool is usually turned at constant speed by a belt drive. The cable is thick and as the spool increases in diameter the belt is permitted to slip when the cable is under sufficient tension. This may wind the cable too tightly, and require maintenance of the belt drive. With photoelectric control, the light beam is projected at the electric eye just above the lag of cable maintained between the capstan and the spool. When winding speed increases, slack is taken up, pulling the cable above the light beam. This places resistance in series with motor field, slowing down the motor until the cable once again interrupts the light beam. At this point, spool resumes winding at full speed, since motor field resistance cuts out.

In testing cable insulation for high voltages, the end of the cable may be run into the test room. The spool itself, remaining outside, can be surrounded by a light beam fence so that any person approaching the spool will disconnect the high voltage line.

In steel mills and in the automotive field, the standard photoelectric unit can control hot objects and freshly painted units, automatic conveyors, automatic door operation, accurate grading by height or size, and similar applications.

In the textile field, the electric eye detects thread breaks and provides stop-motion control on high speed warpers and winders. The older electrical stop motion was rendered inoperative by lint and dust; this is, of course, not the case



Photocell circuit is normally closed and solenoid is normally energized. When light beam is interrupted, solenoid is de-energized and the spring throws the latch into place under the clutch lever

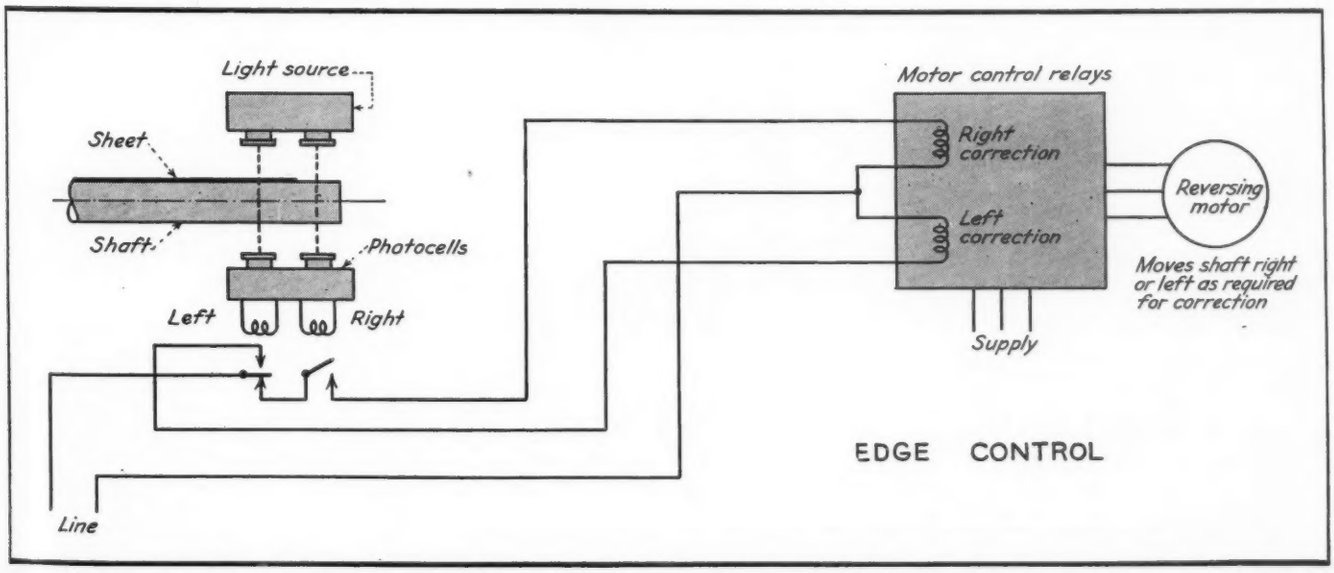
with the electric eye control which is completely positive in operation. A light foil target is suspended from each thread by a porcelain hook. These targets are in line, and one light beam can be projected just beneath them all. Targets are prevented from twisting by a simple wire guide. When a guide breaks, its target drops, breaking the beam and stopping the machine.

In textile finishing, cloth is guided as it runs through processing baths and dryers by pins or clamps engaging its edges. The guides along which these pins travel are moved in and out by reversible motors. Light feelers have been used to control these motors. The feelers contact the edge of the cloth and move the guides to accommodate variations in cloth width. For very light mesh or cloth which is delicate when wet, the feelers require wrinkling of cloth edge before they will make the desired contact. With a lens system projecting a cone shaped light beam, photoelectric control gives better control of mesh and netting than the metal feeler. Breaking the light beam requires no pressure of the cloth. More positive and accurate motor control is obtained. Cloth as thin as mosquito netting can cut off sufficient light for satisfactory operation.

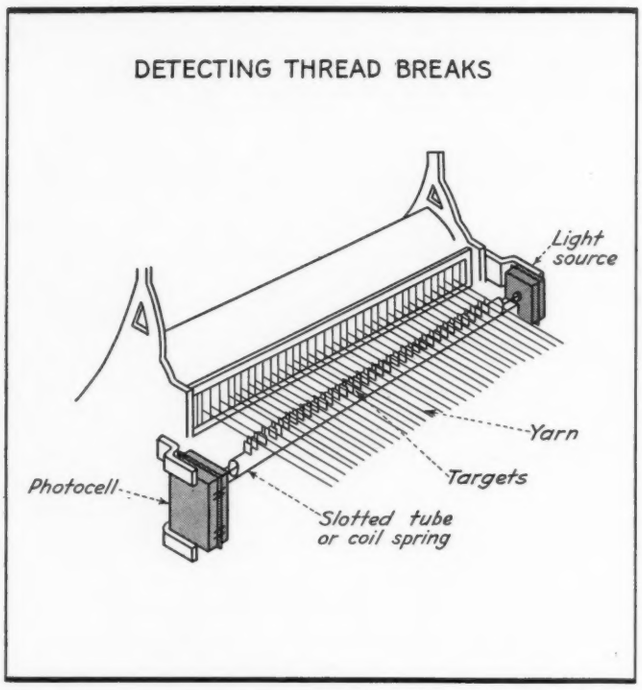
Other applications in the textile industry include speed compensation on finishing machines in the same manner as related in connection with cable winding in the wire industry, and counting of piece goods moving from dryers or conveyors.

For paper break detection, the control beam is projected perpendicularly or diagonally at the paper being processed. In paper processing for edge control the control is used as described in connection with the textile field.

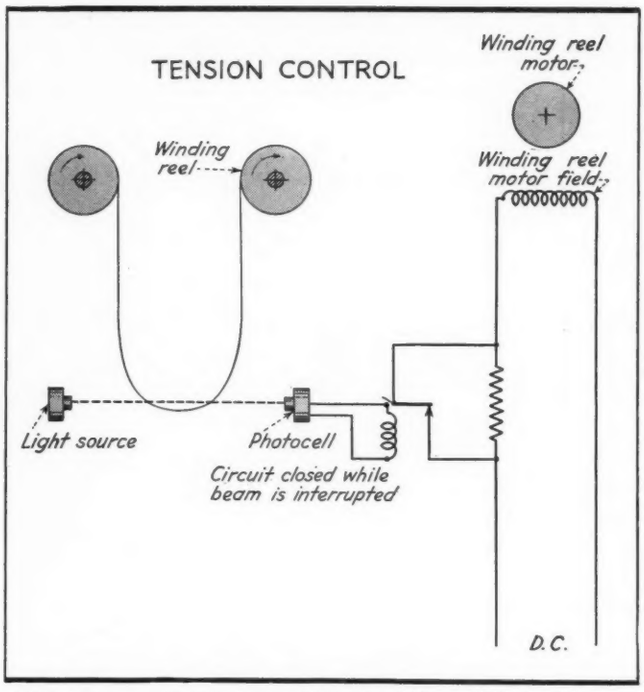
Some photoelectric equipment is offered with an electric counter built into the control, making a compact, easily installed unit. Photoelectric counters, in general, do not compete with mechanical counters. With regularly shaped solid



Sheet is shown in neutral zone requiring no edge correction. When the sheet moves too far to the right it interrupts the right-hand light beam, closing the photocell relay and energizing the reversing motor to cause "right" correction



Light foil targets suspended from each thread by a porcelain hook. Targets are in line and light beam projected beneath



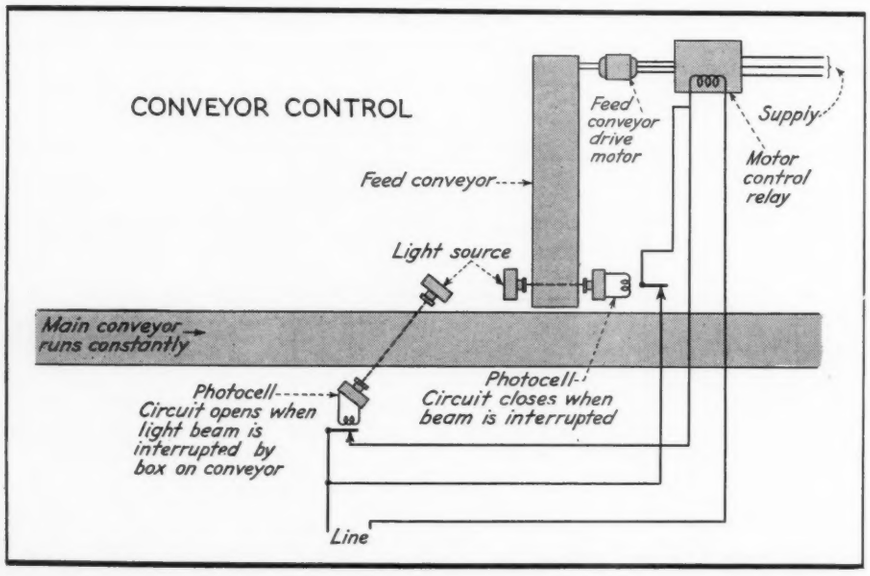
Light beam projected above cable lag. When slack is taken up, motor slows down until beam is again interrupted

objects, and where output of machines may be directly counted by the machine's motions, and where rejections are negligible, mechanical counters are more satisfactory.

The electric eye will usually do a counting job more accurately and economically when actual output, after rejections, is to be counted. In addition, many objects are too light to trip a mechanical counter. With objects of varying or irregular shape, or freshly painted articles, or metal at high temperatures, photoelectric counting is the only solution. It should be borne in mind that a mechanical counter is generally suitable for only one special installation, whereas the electric eye counter is universal in its possible adaptations.

In making a count of such articles as cylinders moving along a conveyor, it may appear that the articles seem too close together to permit light to pass between. An efficient photoelectric control requires very little light for operation. Also there is generally some point on the conveyor where corners or edges of cases must separate such as at the outside of a curve, or at any change in slope. A dead roller on a roller conveyor will automatically space the cases. The rear bottom edges of the cases may be counted at any vertical step, such as when passing from a conveyor to a chute.

Photoelectric controls need no longer be considered in the class of either laboratory or tailor-made equipment. They are available today as standard equipment, and may be as readily purchased as other electrical controls. They have long life under exacting industrial conditions. Maintenance is a negligible factor; well engineered controls require only vacuum tube replacement once a year.



Arrangement which operates feed conveyor only when beam on main conveyor is clear, so interlocked that if a box has begun transfer from feed to main conveyor, feed conveyor continues to operate until transfer is complete

Choice of Equipment

In selecting photo electric controls for a particular application, the following should be determined:

(a) What is the required distance between light source and receiver?

(b) Is the control circuit to be closed when the light beam is striking the electric eye or when the light beam is interrupted?

(c) Is it desirable to close the control circuit in case of current failure or when tube replacement is necessary?

(d) Is operation required on small changes of light or on complete interruption of the light beam?

(e) What are conditions of local illumination?

PATENTABILITY OF MATERIALS

When Invention Enters Into Their Application

H. A. TOULMIN, JR.
Toulmin & Toulmin

THE art of supplying new materials for engineers has been revolutionized in the last decade. Along with the advent of plastics, new alloys, stainless steels, new electroplating coatings and electrocoloring, has also come styling—another weapon for the industrial designing engineer to use in meeting competition. The resourcefulness of the designer is often in direct proportion to his knowledge and skill in the adaptation of his product to new materials. After all of the research and engineering is completed, the question arises whether this investment will be wasted, because a competitor can take advantage of the pioneering that has been done without the expense, and produce the same product at a cut price.

Consequently, the designer turns to patents as his protection for the investment. The question immediately arises whether it is possible to protect the new design because of the designer's originality in the selection and adaptation of new materials to an old purpose.

Most of these questions on patentability due to selection of new materials are difficult ones to determine unless the rules are thoroughly understood.

Generally speaking, when a raw material, or a semi-fabricated or completed new material, is found to have a special quality or characteristic in its new location in the new design, which quality or characteristic had not been observed in it when it was employed in other connections, then this is invention.

Glass as a Typical Example

Glass is a typical material out of which many new qualities have been discovered during the last decade. The art of drawing fibers and weaving textiles of glass has produced sheet and insulating materials, clothing, and a number of new products which are patentable *per se*. It was discovered that the use of woven fiber made out of glass has merit as an insulating cover for electric wire and cable. These fibers, a few ten-thousandths of an inch in diameter, have a tensile strength sometimes as great as a million pounds per square inch, greater than that of steel.

As one Court said:

" . . . Appellant conceived and designed a hollow glass brick with a side, or sides, fluted or ribbed to form prisms, which prisms were given the dimensions and slants essential, under the physical laws of light refraction and light reflection, to catch downwardly slanting rays of light and deflect such rays in an upwardly slanting direction. He also conceived and designed a building to be constructed of such bricks, the ceiling of which building should catch the upwardly deflected rays of light and redistribute them downward for illumination of the interior of the building."

The Court had before it the question:

" . . . whether applicant's work represents invention over the prior art cited plus the common practice of diffusing rays of artificial light downward after their reflection upwardly to the ceiling.

"It is our conclusion that it did involve invention. It seems to us that the concept of causing deflection of light rays in the manner which has been described was inventive. To consummate the conception and embody it in structural form must have required calculations based upon the physical laws relating to light, and, finally, experiments to determine whether the result desired had been obtained."

Taxicab Safety Glass

Such a discovery in itself constitutes invention. Take the case of safety glass windows that are used entirely for taxicabs. These are not double sheets with plastic layers between them, but a single sheet which will withstand four or five times as hard a blow as ordinary glass of the same thickness. When this glass breaks, it goes into fragments the size of the end of your finger, rather than into sharp, jagged splinters.

It was discovered that sheets which have been heated and then suspended before a multitude of fine air jets are tempered because the surface layer of the glass is thus thrown into compression and the interior under tension. The tensile strength of the sheet is increased by the amount of the surface compression. This is a force tending to stretch the glass which must first overcome such compression. It is tensile strength which ordinarily determines whether or not glass will break. The discovery of this new property of glass is invention.

A further good example of discovery of a new property in materials, which is patentable, was that in connection with the manufacture of electrical contacts. A tungsten disk was welded to a steel shank. A tray, holding many small steel shanks, was maintained in a horizontal plane with a thin disk of copper on top of each steel shank. Then a tungsten disk was placed on top of the copper. The whole was placed in a furnace with a hydrogen atmosphere while the copper melted, after which the assembly was removed to a cooler zone, so that the copper could solidify.

Discover Wetting Property

In the beginning, great care was used by the workmen to align the three parts so that the tungsten would be flush all around with the steel shank. Later it was discovered that the surface tension forces of the liquid copper properly aligned the tungsten disk no matter how badly off center. This was due to the discovery that the tendency of copper in a hydrogen atmosphere is to wet both the tungsten and the steel, together with the nearly frictionless film of liquid copper on which the tungsten was free to move.

The result was that it was possible for operators to fix the trays at a faster rate. The tungsten disks, as soon as the copper began to melt, would dart into place, some moving in one direction and others another, but always centering themselves as if they were under accurate physical control. The discovery of this wetting function of an old metal, such as copper, was invention.

Another excellent example of the same property of copper after it was once discovered was in connection with powdered tungsten briquettes that were moderately heated and then placed in a hydrogen atmosphere in a bath of molten copper. They were not completely immersed, so that the capillary forces caused the copper to fill the pores of the tungsten briquettes even above the level of the liquid copper.

There are many latent industries that are awaiting creation due to the delay in the application of physical properties of materials such as metals, now known only as laboratory curiosities. There is an inspiring group of new materials, and new uses awaiting exploration. They are all known in the laboratory.

The development of the carrier wave principle provided the impetus to utilize

electronic emission in radio and in the transmission of many telephone messages over a single circuit. For instance, a thread of lead, if maintained at a temperature near absolute zero, could carry all of the electrical energy available at Niagara Falls all the way to San Francisco without appreciable loss. This property of lead, tin, mercury and some other metals, called supraconductivity, is one that has scarcely been explored.

When practical commercial developments are made, taking advantage of this property, that will constitute invention. Take one of the new alloys like Invar. This alloy has no appreciable thermal expansion within an important temperature range. It was invention to discover the metallic resistant wires, like manganin, with no, or even a slight negative, temperature coefficient of electric resistivity. A discovery upsetting the orthodox physical conceptions of metal is the thing that illustrates what is invention in discovering a new property.

Combination of Old Materials

Take the case of new combinations of old materials which form a new product. Over in Germany it was discovered that wood could be immersed in molten nonferrous metals, absorbing those metals, and forming a new type of machinable product. Then there came into the field a new plastic made from lignin, a product of the refinement of cellulose. Wall boards and other products are made from this material. Then there was the combination of sponge rubber and pig hair that made a new upholstering, and wool made artificially from casein and dresses from glass fabric.

There are three aspects of this problem of what constitutes invention in connection with materials: "When we find that a material has a new quality, it is invention; when we discover a new usefulness and adaptability of a material, that is invention; when materials take on new forms not heretofore known, that is invention; and when we have a new combination of materials, that, again, is invention."

For instance, let us take the new synthetic rubbers, such as Thiokol, Neoprene and Buna. Here are a few of the synthetic rubbers with startling new uses and adaptabilities, both alone and when mixed with natural rubber.

Take the case of selecting such materials as the polymers made from lime and coal. This new material was applied in a thin layer to the surface of a printing roller made of rubber. The claim read somewhat like this, which was allowed by the Patent Office:

"1. A composite roller comprising a pervious resilient rubber body having an integrally attached synthetic rubber covering, said covering being sufficiently resilient to permit flexing of the rubber body and so thin as to be relatively incompressible, smooth and impervious in contrast to the resilient pervious body."

It was discovered that this synthetic rubber not only had the quality of being resistant to oils, greases and wash-up liquids, but that it also developed the unexpected property of having a hard, smooth surface, so that it would provide the ideal printing surface. By making it sufficiently thin, its lack of resiliency and flexibility necessary in a printing roller was made up by mounting it upon a relatively thick, very resilient, rubber base, which, if not so protected, would be attacked by the oils, greases and wash-up liquids. Here was a true example of invention in applying a new material of certain known qualities to an old use of a printing roller, but in a new mechanical combination and with the development of new properties from the synthetic rubber due to the adaptation of it to a very special purpose having a number of unusual requirements.

When a new material placed in an old construction develops a new use or develops new qualities, that are not obvious, then it is invention. As a Philadelphia Court said:

"It was known by a few persons that copper sulphate would absorb ammonia gas, but the knowledge of that abstract chemical fact, which no one used, is quite different from the disclosures of the patent. The patentees translated a more or less isolated and hitherto comparatively unimportant fact of chemistry into a useful industry. That the process came immediately into general use and has replaced all other like processes is persuasive of invention."

Speed Regulator Invention

Another good example, in a practical way, of this principle comes to mind. A few years ago, an inventor discovered that if he placed a series of iron cores in coils and moved them in and out of the coils, an old electrical device in itself, but hooked up this arrangement in a three-phase circuit of an alternating current motor, that he was able to control the speed of that motor from substantially zero to full speed without undue heat loss. He could not only control the speed of the motor, but could control the amount of current used in proportion to the speed of the motor. This opened up a vast new field in speed regulation, particularly with reference to ventilating fans and similar mechanism. Here was an adaptation of old materials and an old principle to an entirely new use which was patentable.

A San Francisco Federal Court said:

"It is true that the characteristic of vulcanized rubber has long been known, that the skidding automobile demonstrated the fact that with water lubrication on the highway the rubber of an automobile tire supporting a heaving automobile showed little frictional resistance. The patent office, however, decided that the use of water lubricated soft rubber to support a shaft constitutes invention.

"The real problem of using soft rubber in the bearing or on a journal results from the intimate contact with the metal which it supports. This contact leaves no space for a lubricant on the surface which supports the weight. Sherwood believed that he could force water between the rubber and the steel and thus make practical application of the desirable qualities of rubber in the structure. We agree with the trial court and with the patent office that this constituted invention and that patent No. 1,416,988 is valid as against the claim of no invention."

"Let Us Assume . . ."

Let us illustrate further with one of the new materials that is a plastic. If a designer should discover that in adapting a plastic handle to a machine that it was possible to print instructions on the handle, a quality which a plastic had not been known to have prior to that time, this would be a patentable discovery. Or take the case of oil in a hydraulic circuit. Let us assume that it was discovered in carrying the oil up to very high pressure that it developed properties that it had never shown ordinarily for lubricating or hydraulic pressure purposes. This would also be patentable. If a motor armature were cast in a synthetic resin under pressure in order to provide the necessary insulation and it is discovered that this insulating plastic had the property of acting as a condenser, or the property of storing electrical energy, then this newly discovered property would be patentable.

Then, there is another angle to this question that is of interest. An inventor discovered that by mounting a thin sheet of expensive stainless steel on a cheap carbon steel support and thereafter bending both the thin sheet of stainless steel and the cheap carbon steel support in an arc, that it was possible to stretch the stainless steel sheet so as to cause it to lock on the carbon steel support due to the tension under which the stainless steel sheet was put. This was broadly new as the inventor utilized the tensile strength and resiliency of the thin steel sheet to lock it on the carbon steel base for use on automobile tires. The use of solid stainless steel was, of course, too expensive.

Where a machine has acquired new functions and useful properties, it may be patented, even though the only change made in the machine from prior designs has been the substitution of one of its materials by another.

DAMPERS AND SUPPRESSORS–I

For Controlling Torsional Vibrations in Crankshafts

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Detroit Diesel Engine Division of General Motors Corporation

IN multi-cylinder internal combustion engines which operate through a wide speed range it is practically impossible to keep that range free from speeds at which severe crankshaft torsional vibrations will occur. However, since the resonant torsional vibrations of a crankshaft are governed by the conditions of resonance existing between the torque impulses and the natural frequencies of torsional vibration of the crankshaft system, it is sometimes possible to control these conditions to some extent by the design of the crankshaft. This can be easily performed, however, only in engines having a narrow operating speed range.

It is possible to flexibly and frictionally connect the flywheel to the crankshaft so that the system will have little response to the exciting forces at the resonant frequencies. Also, mechanical devices have been used for automatically changing the moment of inertia or the stiffness of the crankshaft at certain engine speeds in order to avoid resonance. Methods have also been developed in which the moment of inertia of the crankshaft is varied through the cycle of engine revolution. Devices such as these, although they have merit, are not used widely, usually because they are too complex and too expensive. The most common method of torsional vibration control is the use of a suppressor of some sort to eliminate the existing torsional vibrations.

The need for a torsional suppressor depends upon several factors. In passenger car design, vehicle vibration and noise are the deciding factors. In bus and truck engine design, crankshaft stress, timing gear wear, and camshaft operation may be the important factors. In motorship drives, crankshaft and propeller shaft stresses, auxiliary and main gear drive, wear and noise are usually the important items determining the need for the application of a suppressor. Each design of engine, together with the driven equipment, must be investigated separately in order to select the proper type of suppressor.

Torsional vibration suppressors as commonly used fall into four general classes: dampers, tuners, tuners with damping, and de-tuners.

Dampers That Introduce Work Absorbing Forces Into System

These introduce work absorbing forces which are brought into operation when the crankshaft torsional vibration amplitude exceeds a predetermined amount. In their essentials, dampers consist of heavy flywheel rings mounted coaxially with the engine crankshaft and caused to rotate by it through friction, either solid or fluid.

When the crankshaft speed of rotation is uniform, the shaft and the damper flywheel rotate as a unit. During crankshaft torsional vibration, the flywheel has the same oscillation only if the torque necessary to give the flywheel maximum acceleration is less than that causing the friction surfaces to slip. If the amplitude of vibration of the crankshaft, at the damper location, is caused by a greater torque than this, slip between the two surfaces occurs with consequent energy loss. Through correct design the energy absorption by friction balances the harmonic torque input at any desired amplitude thus limiting the vibration stresses at critical speeds to safe values.

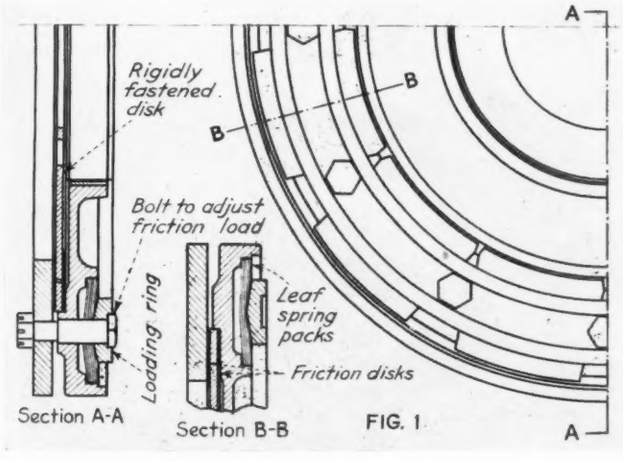


FIG. 1

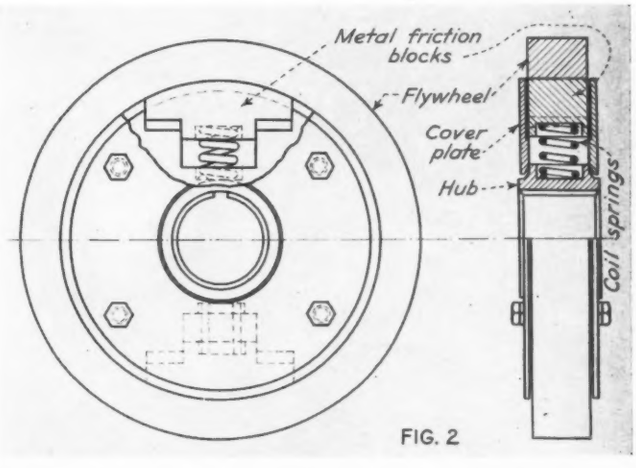


FIG. 2

Torsional vibration damper shown in Fig. 1 is inclosed in a housing and lubricated, bronze friction material being used. The assembly is bolted to the crankshaft, the disk being the rigidly fastened member. The flywheel is driven by the crankshaft through the friction disks which are loaded by leaf spring packs through a loading ring which is also a part of the flywheel mass. Frictional load is adjusted by means of a nut and bolt. The unit is assembled to the front end of the crankshaft.

Damper in which the frictional torque setting varies as the square of the engine speed, this design shown in Fig. 2 incorporates a damper flywheel against which metal friction blocks bear. The blocks are given an initial load against the inner periphery of the flywheel by the coil springs. The effect of centrifugal forces upon the blocks provides an automatic increase of the friction torque setting. Freedom from criticals over the entire speed range is obtained. (*Continued on next page*)

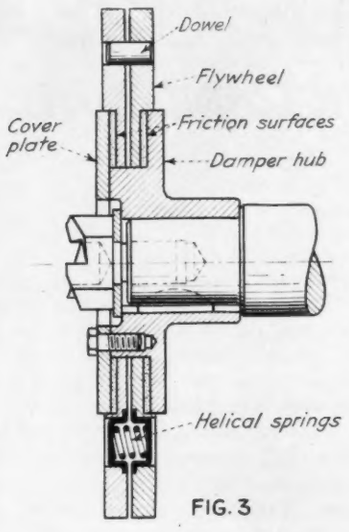


FIG. 3

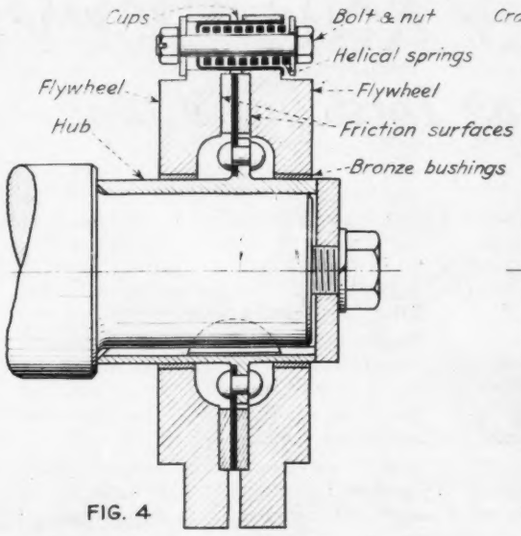


FIG. 4

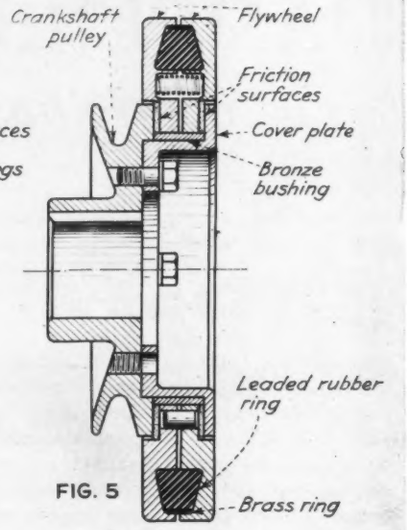


FIG. 5

Typical torsional vibration damper, shown in Fig. 3, is used widely in automotive type internal combustion engines. The damper is keyed to the front end of the crankshaft, where the amplitudes of torsional vibration are the greatest, through the damper hub. The flywheel, or inertia mass, is connected to the crankshaft by means of friction material such as brake lining or clutch facing material. In some instances, if the damper can be inclosed and lubricated, metallic friction materials are used. The friction surfaces are loaded by helical springs which force the two halves of the flywheel mass against the friction material. The springs are designed to have the lowest possible rate of action in order to minimize the reduction of spring load which may occur with wear of the friction surfaces. The cover plate fastened to the hub, holds the assembly together and acts as an additional friction surface. The two halves of the flywheel mass are doweled together to prevent relative motion.

Vibration damper, shown in Fig. 4, is similar to the type shown in Fig. 3, except that it is possible to adjust the spring load to compensate for wear on the faces of the friction material. Again the damper is keyed to the front end of the crankshaft through the damper hub. The two halves of the damper flywheel, which stores up and gives out inertia forces, are connected to the hub through the friction surfaces which are loaded by helical springs. The springs, which are placed in cups around the outside edges of the two halves of the flywheel, are compressed by means of tension bolts and nuts. Adjustment of the load on the friction faces, and consequently of the torque required to cause flywheel slip, can be accomplished by use of this method of construction when properly designed. To pilot the damper flywheel on the hub, preventing "run-out," bronze bushings are pressed into the bore of each half of the flywheel. The bushings also serve as bearings when slip occurs between flywheel and hub.

Damper for high-speed engines, shown in Fig. 5, in which the friction torque required for full range damper effectiveness varies approximately as the square of the engine speed. Increased frictional torque required to slip the damper flywheel in relation to the crankshaft is obtained from the action of centrifugal force upon a "leaded" rubber ring. The specific gravity of this ring is increased to about 4.7 through the addition of litharge to the rubber. As the engine speed increases the ring causes a wedging action upon the two halves of the damper flywheel causing them to bear with increased force against the friction surfaces. A brass ring surrounds and confines the outer rim of the rubber thus preventing any "flow" because of the force acting upon it. The two halves of the flywheel are guided on the cover plate by a bronze bushing which acts as a bearing. The cover plate, which also acts as one of the fixed friction surfaces, is bolted to the crankshaft pulley which is keyed rigidly to the crankshaft.

Tuners That Operate by Neutralizing Primary Torques

Dampers as has been described operate by dissipating energy as it is passed into the crankshaft system by exciting harmonic torques. The same result can be obtained by adding another vibrating system to the main one which opposes the exciting torques with a vibratory torque.

In this type or group known as tuners there is no energy dissipation as occurs in dampers, but rather a neutralization of the torques which might possibly lead to serious vibration. Three examples are shown in the following illustrations of this type of torsional suppressor which consist essentially of a heavy flywheel connected elastically to the crankshaft at a position or place remote from the node of vibration.

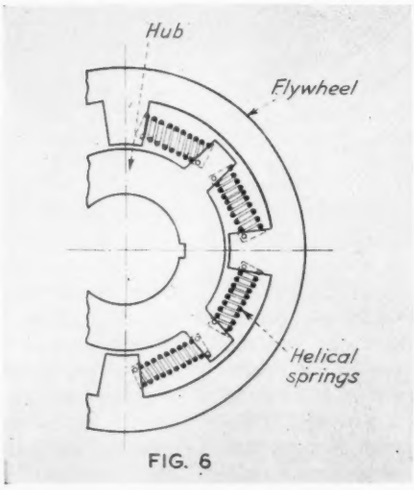


FIG. 6

Common type of purely tuned crankshaft torsional vibration absorber is shown in Fig. 6. As may be seen from the illustration, there is no provision for the dissipation of vibration energy through friction as in the foregoing types. This ring type tuner consists of a flywheel mass coupled through a number of helical springs to a spider arm hub which is keyed to the crankshaft at the end remote from the flywheel. In the design of these units it is necessary that the springs be properly proportioned to transmit the maximum torque without being too highly stressed. The flywheel and spider units are designed and the masses so distributed that the natural frequency of the tuner is made the same as that of the crankshaft system.

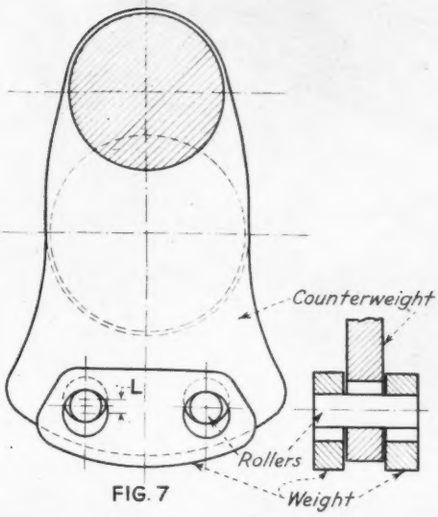


FIG. 7

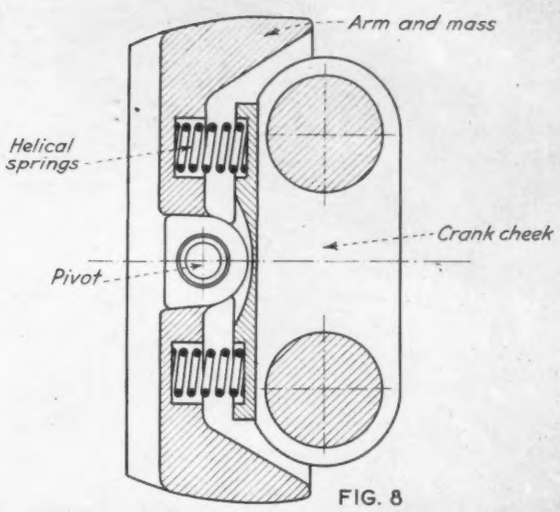


FIG. 8

Torsional vibration suppressor shown in Fig. 7 is a form which is being used to some extent on aircraft engines. It consists of a weight suspended on two rollers which are a free fit in the weight and in the crankshaft counterweight. Means for keeping the weights on the rollers are also used. This design is acted upon by centrifugal forces in the same manner as gravity acts upon a simple pendulum. It thus has a natural frequency of vibration which is proportional to engine speed when vibrating through small amplitudes.

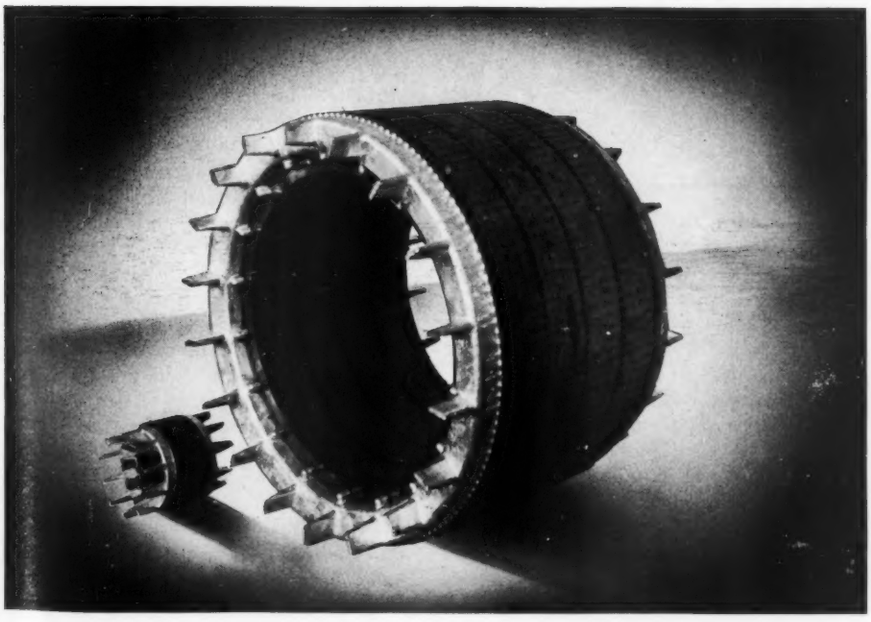
The mass acts as a simple pendulum which swings through an arc of radius *L*. The natural frequency of the unit can be made to respond to any harmonic of the crankshaft throughout the speed range. The pendulum acts in the same manner as the tuners described in Fig. 6 at the crankshaft harmonic to which it is tuned.

The effectiveness of this type depends upon its amplitude of oscillation, the effective radius, the inertia of the weight, and the tuning or natural frequency. Since the effectiveness does vary as the square of the speed it is possible to use light weight tuners for vibrations occurring at high speeds indicating one reason for its incorporation in aircraft engines. The rollers are so located as to lie close to the center of gravity of the pendulum weight and far enough apart so that the weight will not be lifted from them by inertia forces thus causing rattling and loss of vibration absorbing effectiveness.

Simple type of purely tuned crankshaft torsional vibration absorber illustrated in Fig. 8, although of different form is similar in action to that shown in Fig. 6. The form shown in Fig. 8 consists of an arm and mass pivoted to a plate attached to a cheek of the crankshaft. The angular position of the mass in relation to the cheek is fixed by helical springs. In the design of these units it is also necessary that the springs be properly proportioned to transmit the maximum torque to the mass and arms without being too highly stressed.

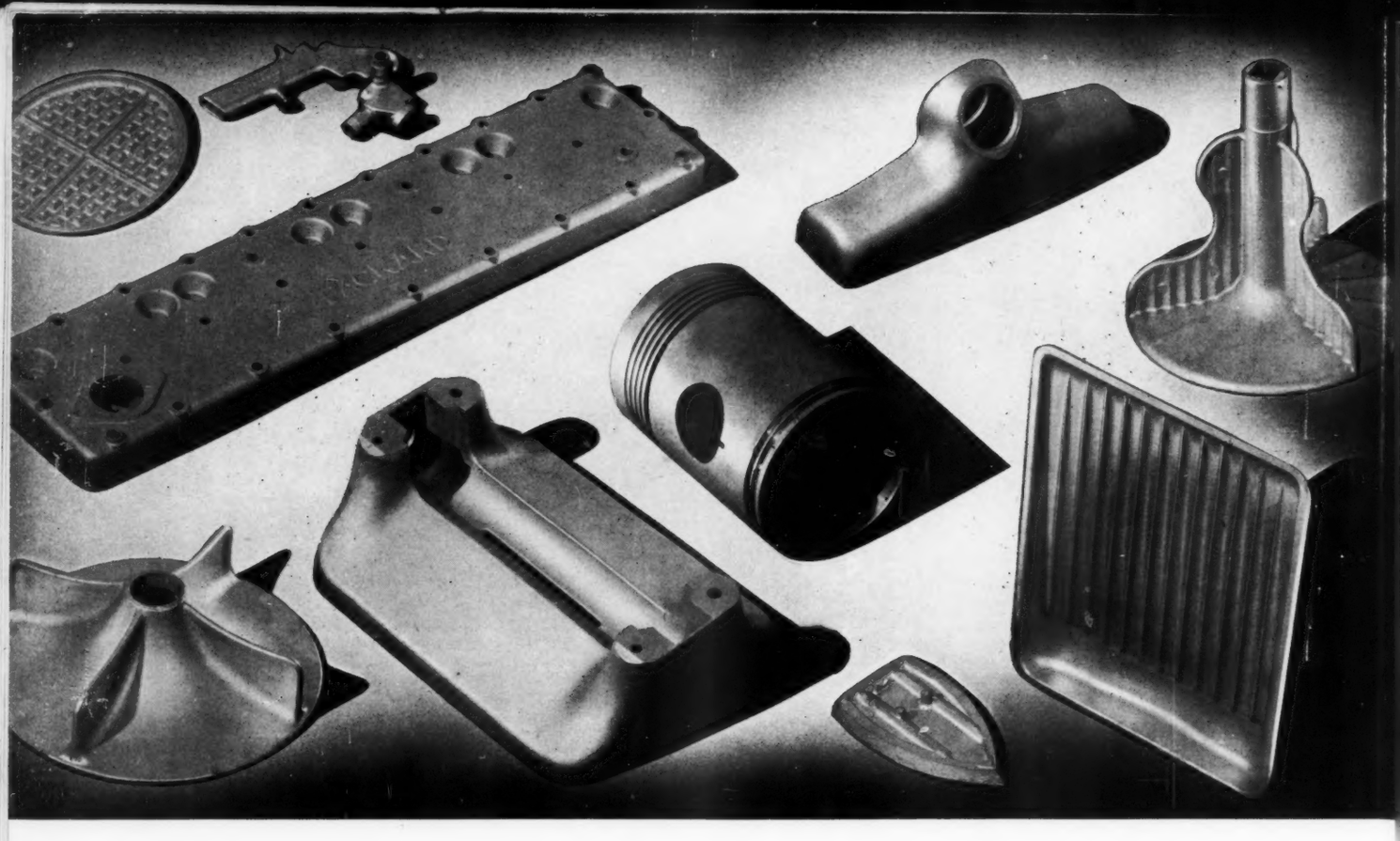
EDITOR'S NOTE. More torsional crankshaft vibration suppressors of the types which are classified as tuners with damping and de-tuners will be described and illustrated in a future number.

Pressure-Cast Aluminum Rotors



Coefficient of expansion and contraction of aluminum is about twice that of steel yet cast aluminum rotors with steel cores are made in sizes up to 20 in. diam. by the Reliance Electric and Engineering Company. By preheating the steel core to about 600 deg. F. before casting the aluminum around the core, by using high casting pressures and by speeding up the casting press the effects caused by the difference in rates of contraction have been reduced. Advantages claimed for the cast type of construction are concerned with fatigue properties against vibration, degree of balance, ventilation and electrical characteristics.

Cast-type rotors are now operating under such exacting conditions as are encountered in reversing drives for roller and catcher tables, mining machinery drives and metal drawing presses. The small rotor in the illustration is for a 1 hp., 1,200 r.p.m. motor, the large rotor is for a 250 hp., 1,200 r.p.m. motor.



ALUMINUM ALLOY CASTINGS

Design Possibilities and Range of Applications

WHEN SPECIFYING that a part be cast in aluminum engineers should be governed by many factors before making a selection of the alloy that will best suit the loads and conditions to which the casting will be subjected in the intended service.

At present aluminum alloys are available to suit almost every condition. Some alloys are corrosion resistant to salt water attacks; others have hardness properties and wearing qualities suitable for bearing surfaces. Alloys can be supplied to meet conditions where weight is a factor in the final unit; or where strength is required with a certain amount of ductility; or where machinability is a factor; and to meet numerous other conditions that may affect the final use of the casting. However, all the conditions cannot be met just through the selection of a certain aluminum alloy alone, as some thought must be given to the heat-treatment that will be required to obtain properties determined by the final requirements and use of the casting.

Aluminum alloy castings are produced in various ways. One method is by casting in sand where either dry or green sand cores form the interior or exterior of the casting. No doubt this process is familiar to all. All aluminum alloys can be used to produce castings by this method, they can also be heat-treated to obtain certain physical and hardness properties. This method of casting is used when the design is so intricate that it cannot be produced by another method; or possibly when a small quantity of castings are required for test units before approving the final design; or possibly the amount required does not warrant the purchase of the equipment necessary for the other methods of manufacture. Very satisfactory parts are produced by sand casting, which are readily accepted by designers and engineers.

THIS ARTICLE was prepared from materials furnished by the Permanent Mold Casting Division of the Aluminum Association. The tables of properties and composition of aluminum alloys were taken from A.S.T.M. "Tentative Specifications for Aluminum-Base Alloy Permanent Mold Castings."

Another method of manufacture is the semi-permanent mold process, which consists of using a dry sand core for the interior and a metal mold for the exterior surfaces. This method is used when it is impossible to remove steel or iron cores from the interior during the casting process, and also when a much smoother surface is required on the exterior than can be obtained in a sand casting. All aluminum alloys can also be used in this process, and castings can be heat-treated to obtain the required or necessary physical and hardness properties.

A good example of castings made by the semi-permanent mold process is the aluminum automotive cylinder head, in which the exterior surfaces and combustion chambers are cast to size in a metal mold, while the interior water chambers are cast with dry sand cores. These dry sand cores are removed after the casting has been taken from the mold. The semi-permanent mold process is also acceptable and approved by de-

signers and engineers, but the initial cost of the equipment is somewhat more costly than the pattern equipment for sand casting, however, a much longer equipment life is realized, and because of higher production rates a reduction in the piece or pound cost is possible.

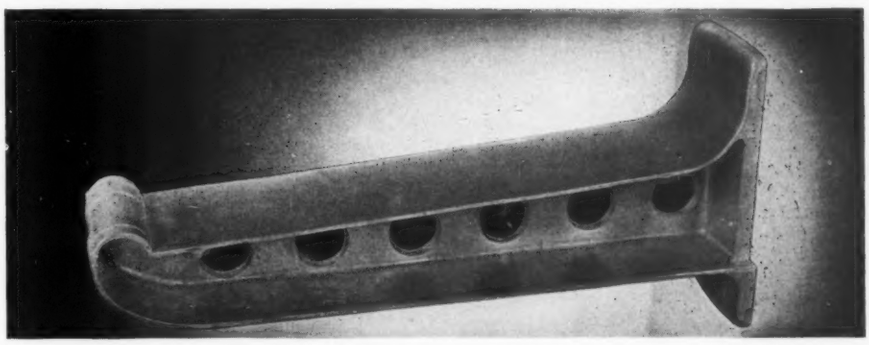
For several years there has been a growing tendency to use permanent mold aluminum castings where production and quality castings are required. In this process removable steel or iron cores are used for the interior surfaces and metal molds for the exterior surfaces. However, this process is dependent upon a means of suitable core removal from the casting, either while still in the mold or after the casting has been removed from the mold. By this process closer casting limits, weights and dimensions can be maintained almost indefinitely. A much finer grain structure, both on the interior and exterior surfaces, is obtained because of the chill casting effect which naturally reflects in the final physical and hardness properties obtained. Smooth interior and exterior surfaces are obtained through this method of manufacture, which are sometimes highly desirable for the finished appearance of the item. All aluminum alloys can be used in this process and when given the required heat treatment, excellent properties are obtained.

By the permanent mold method of manufacture the sections of the castings are maintained more uniform, which does much to keep the weight of the finished product within specified limits. Casting limits of ±0.010 in. can be maintained. However, this is dependent upon the shape and size of casting. Sections of 0.090 to 0.100 in. are being satisfactorily cast; however, this also is dependent upon the shape and size of the casting. Chill cast holes may be had; however, this is dependent upon their diameter and depth. Minimum draft of approximately 0.010 in. per in. is required, which is very slight in comparison to sand cast pattern equipment.

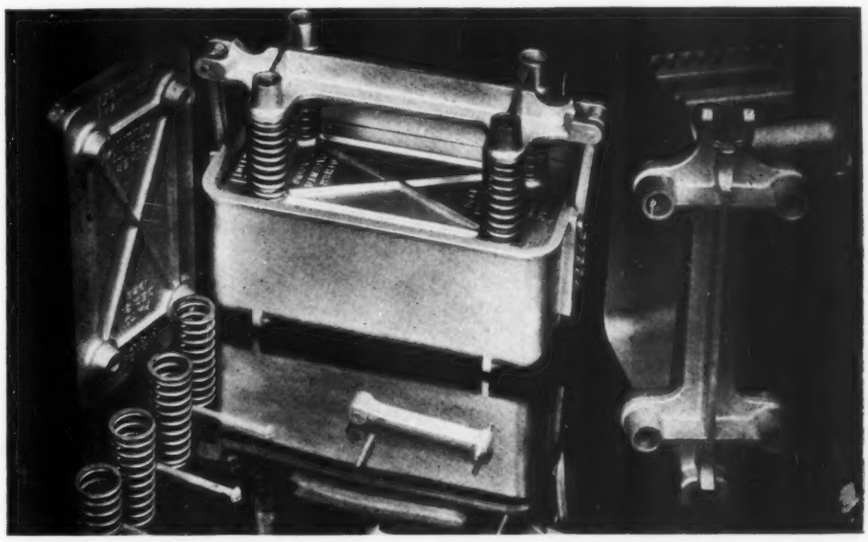
Some common castings of the permanent mold type are internal combustion engine pistons, steak platters, connecting rods, waffle iron plates, washing machine agitators and parts, cylinder heads, golf cups, street markers, instrument cases and housings, and master cylinder hydraulic brake pistons. Other applications are numerous. Castings can be supplied for parts where final weight is a factor, and for many other requirements for the finished article, such as appearance, finish, and corrosion resistance. Castings weighing from a fraction of an ounce to more than 100 lb. are being produced. While the initial cost of the necessary equipment is somewhat higher than the equipment required by the other two processes of casting, a much longer equipment life is effected

Physical Properties of Aluminum-Base Alloy Permanent Mold Castings

Alloy	Condition	Tensile Strength Min. Lb. per Sq. In. (To the nearest 1,000 lb. per sq. in.)	Elongation In 2 In. Min. Per Cent (To the nearest 1 per cent)
1	Heat treatment No. 1	33,000	4.5
1	Heat treatment No. 2	42,000	1
1A	Heat treatment No. 1	33,000	4.5
1A	Heat treatment No. 2	42,000	1
2	As-cast	23,000	....
3	As-cast	23,000	....
4	Heat treatment No. 1	28,000	....
4	Heat treatment No. 2	38,000	....
5	As-cast	26,000	....
6	Heat treatment No. 1	28,000	....
6	Heat treatment No. 2	36,000	....
7	Heat treatment No. 1	32,000	....
7	Heat treatment No. 2	36,000	....
8	As-cast	27,000	3
9	As-cast	21,000	2
9A	As-cast	21,000	2.5
10	Heat treatment No. 1	28,000	5
10	Heat treatment No. 2	30,000	3
11	Heat treatment No. 1	33,000	....
11	Heat treatmcnt No 2	40,000	....



Pullman baggage rack bracket made in aluminum by permanent mold casting method



Ham press made of aluminum permanent mold castings. The ham is pressed inside the container which shapes the meat during cooking

and because of a higher production rate a reflection is noted in the piece or pound cost of the casting.

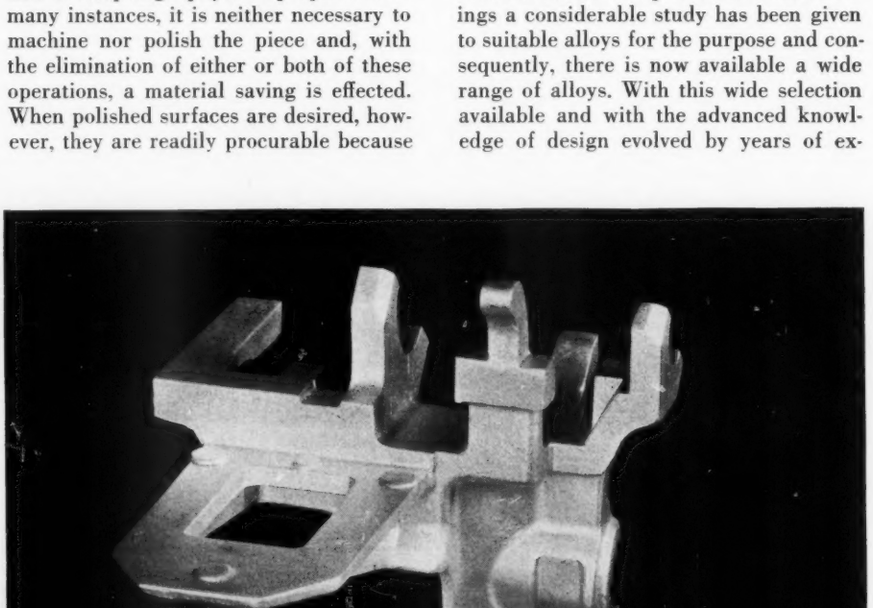
While mention has been made of heat-treated aluminum alloy castings, heat-treating is not always required. However, it is generally recommended should it be necessary to obtain the required physical and hardness properties, to improve the machinability, or to remove internal casting strains and permanent growth of aluminum alloys.

There are two types of heat-treatment: the solution treatment, and the precipitation and solution treatment. Recommendations are made as to the type of heat-treatment required, dependent upon the final use of the article, and upon the type of aluminum alloy selected, to meet the requirements. Heat-treatment is required when a stable part is necessary or where a part may be used at elevated temperatures. When castings are used at elevated temperatures it is absolutely necessary that the permanent growth be removed. Most all of the aluminum alloys are susceptible to heat-treatment and should be given careful consideration in the design and application of the finished product.

With regard to the uses of finished products certain alloys are required for internal combustion engine pistons, while certain other types are required for corrosion resistance, for strength at elevated temperatures, for various co-efficients of expansion that may be required, various weight conditions, appearance and final finishes, and machinability through heat-treatment. There are innumerable uses for aluminum alloy castings and each day new uses and applications are being made throughout industry.

Manufacturers producing assemblies find many reasons why permanent mold castings should be used. In the first place, permanent mold casting is admirably suited to quantity production. Secondly, the resulting castings are of fine grain structure unusually free from dross, air pockets, porosity and irregularities of structure. Permanent mold cast parts are excellently suited for heat-treatment and develop high physical properties. In many instances, it is neither necessary to machine nor polish the piece and, with the elimination of either or both of these operations, a material saving is effected. When polished surfaces are desired, however, they are readily procurable because of the exceedingly close or dense structure of the castings.

Because of the fine grain structure of permanent mold castings, not only are the results obtained from any given alloy generally up to expectations but rejections because of imperfections or irregularities in the castings are at a minimum. Since the advent of permanent mold castings a considerable study has been given to suitable alloys for the purpose and consequently, there is now available a wide range of alloys. With this wide selection available and with the advanced knowledge of design evolved by years of ex-



Aluminum permanent mold casting used as a teletype machine part

Table of Supplementary Data

Alloy	Approximate Specific Gravity	Corrosion Resistance	Casting Properties	Response to Heat Treatment	Principal Use	Machinability
1	2.76	Good	Good	Yes	General, where high strength and fair ductility are required	Good
1A	2.76	Good	Good	Yes	General, where high strength and high ductility are required	Good
2	2.83	Fair	Good	Yes	General, automotive cylinder heads and washing machine agitators	Excellent
3	2.79	Fair	Excellent	No	General	Good
4	2.90	Fair	Good	Yes	Automotive pistons	Excellent
5	2.88	Fair	Good	Yes	General	Good
6	2.68	Good	Fair	Yes	Pistons	Fair
7	2.67	Good	Fair	Yes	Pistons	Fair
8	2.64	Excellent	Excellent	No	General thin sections	Poor
9	2.66	Good	Good	No	General and cooking utensils	Poor
9A	2.66	Excellent	Good	No	General with maximum corrosion resistance	Poor
10	2.70	Excellent	Excellent	Yes	General where high strength and corrosion resistance are required	Good
11	2.78	Good	Fair	Yes	Aircraft and diesel engine pistons	Good

perience, permanent mold castings have proved to be highly satisfactory in the diversified fields to which they have been applied.

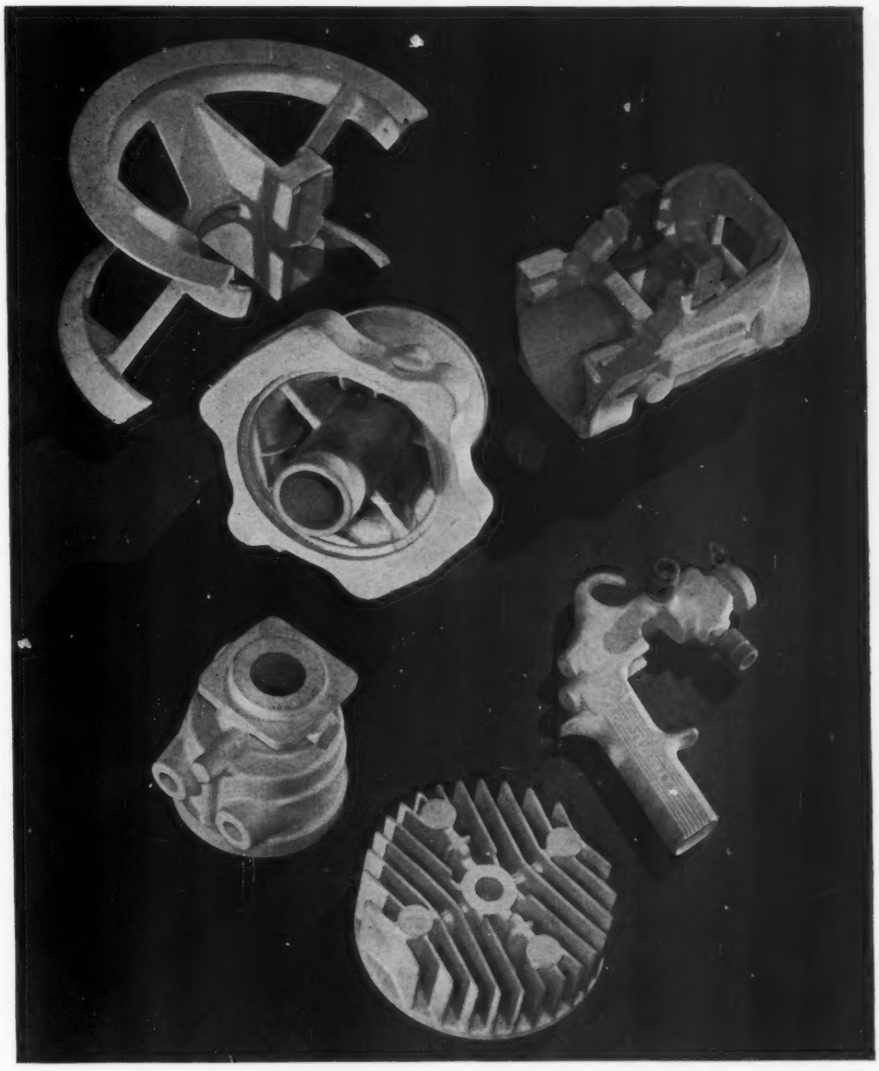
Permanent mold castings can be held to close dimensions. Even the larger castings are capable of being held to reasonable tolerances, such as plus or minus ten thousandths of an inch and, in specific cases, closer tolerances can, and have been maintained in production.

The range of adaptability of the permanent mold process is further widened by the ability to use sand cores where it would be impossible to extract metal cores after the casting has been made. This modification of the process, as has been stated, is known as the semi-permanent mold method, and retains all the major advantages of the full permanent mold process. The design of the part to be produced may dictate which method must be used, but where a choice is permitted experience is the best teacher and an expert permanent mold authority can readily determine which is suitable. Either method, of course, offers the advantage derived from the use of a permanent mold for the exterior surface.

Since a smaller amount of finish is required for permanent mold castings than for sand castings and since it is possible to hold to more exact limits, less metal is required per piece, often resulting in a saving of more than ten per cent. While the initial cost of a permanent mold is usually greater than that of production pattern equipment this factor is off-set to a large extent by the saving in weight realized and the additional saving in lower manufacturing costs.

The basic difference between the permanent mold process and the die casting process is that permanent mold castings are poured in a metal cavity by gravity feed, while in the die casting process the metal is fed into a metal cavity under pressure. In a properly designed permanent mold, the solidification of the metal takes place progressively. The part of the casting farthest removed from the sprue and risers solidifies first and it is this progressive feeding of the casting, together with the rapid chill the metal receives on coming in contact with the metallic mold, that makes possible a casting with a high degree of metallurgical perfection and with properties unique to the permanent mold process.

The variety of forms and shapes that can be handled under present advanced practice is so great that manufacturers are strongly advised to explore this method thoroughly before turning to any other means of producing metal parts. The fine uniformity of results, the ease of machining and polishing, the achievement of lightness with strength, the splendid appearance of permanent mold castings and other advantages are too potent in commercial values to be overlooked.



Group of aluminum permanent and semi-permanent mold castings typical of the wide variety of parts that can be made by this process

Chemical Compositions of Aluminum-Base Alloy Ingots for Permanent Mold Castings

Alloy	% Al	% Cu	% Fe	% Si	% Mg	% Ni	% Zn	% Mn	Other Impurities
1	*	4.5	0.8	2.5	0.2	....	0.2	0.3	0.3
1A	*	4.5	0.8	2.5	0.05	....	0.2	0.3	0.3
2	*	7.0	1.1	3.0	0.1	....	2.5	0.3	1.0
3	*	4.5	0.8	5.5	0.1	....	0.5	0.3	0.5
4	*	10.0	1.3	0.8	0.25	0.3	0.4	0.25	0.3
5	*	10.1	1.0	4.0	0.25	....	0.5	0.3	0.5
6	*	1.0	1.1	12.0	1.1	2.5	0.1	0.05	0.2
7	*	1.5	0.7	12.3	0.6	0.05	0.4	0.75	0.3
8	*	0.2	0.6	11.0	0.05	....	0.3	0.5	0.5
9	*	0.3	0.6	5.0	0.05	....	0.3	0.3	0.3
9A	*	0.1	0.6	5.0	0.05	....	0.05	0.05	0.05
10	*	0.15	0.4	7.0	0.3	....	0.05	0.05	0.05
11	*	4.0	0.8	0.6	1.5	2.0	0.1	0.2	0.2

* Remainder.

MOTOR DRIVES FORVE

SKETCHES of vertical shaft drives show how frame construction, motor mountings, reduction units and bearing units can be adapted to meet requirements of torque, available space, rigidity, operating conditions and service requirements. Although the units shown here are taken from agitator applications, the principles involved are the same as and can be applied to almost any vertical shaft installation. The first part of this article, page 396 of PRODUCT ENGINEERING for September, 1939, presented V-belt and gear-head motor drives.

Compiled by H. D. SEYMOUR
Monsanto Chemical Company

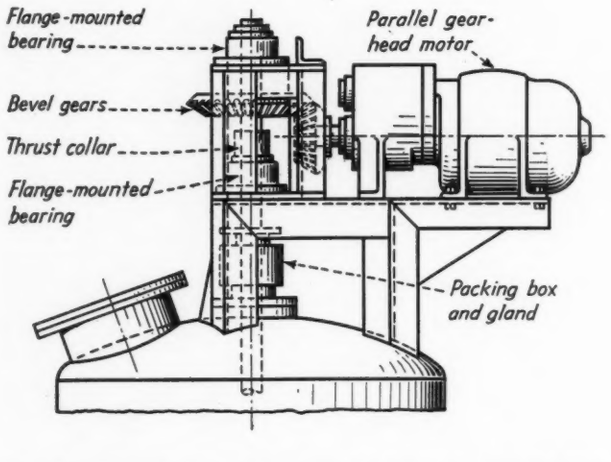


Fig. 1—Transmission is from gear-head motor through bevel gears to shaft in this unit for medium or heavy duty, slow-speed service. Installation is compact, but exposed gears may be objectionable.

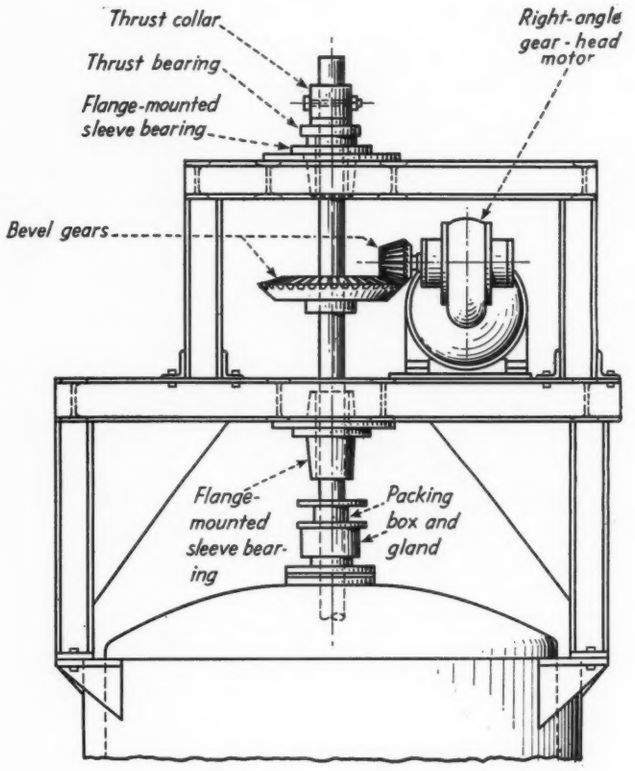


Fig. 2—Right-angle gear-head motor drives vertical shaft through bevel gears in this large, heavy-duty, slow-speed drive. Thrust collar supports shaft weight, and flange-mounted bearings provide rigidity.

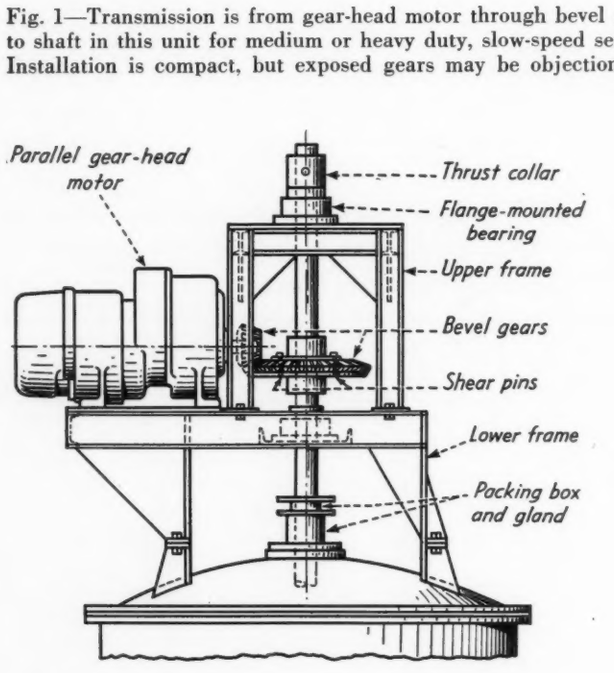


Fig. 3—Shear pins are used in the bevel gear on the vertical shaft of this unit to prevent damage caused by overload or snagging. Drive, as in Fig. 1, is from parallel gear-head motor through bevel gears. In comparing with above, note the spacing of the bearings to increase shaft rigidity.

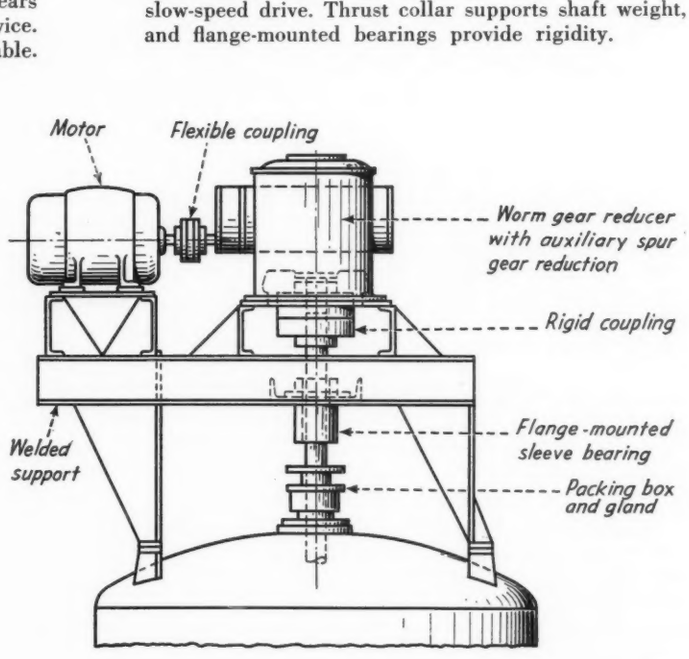


Fig. 4—Slow-speed, medium or heavy-duty drive. Motor is direct-connected to worm gear speed reducer with output shaft coupled directly to vertical shaft. Auxiliary flange-mounted sleeve bearing, suitable for slow speeds, provides additional rigidity. Frame is welded construction.

VERTICAL SHAFTS—II

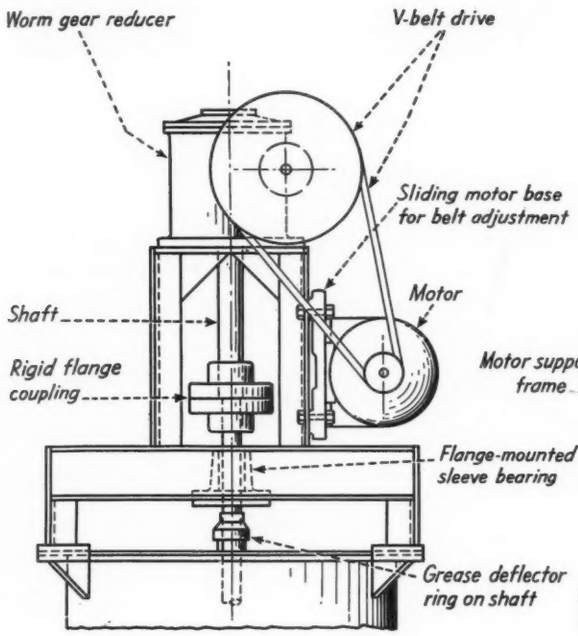


Fig. 5—A more compact installation than that of Fig. 4, the motor in this unit is mounted on the side of the supporting frame. Sliding base for motor, common for this type of mounting, facilitates adjustments of V-belt drive.

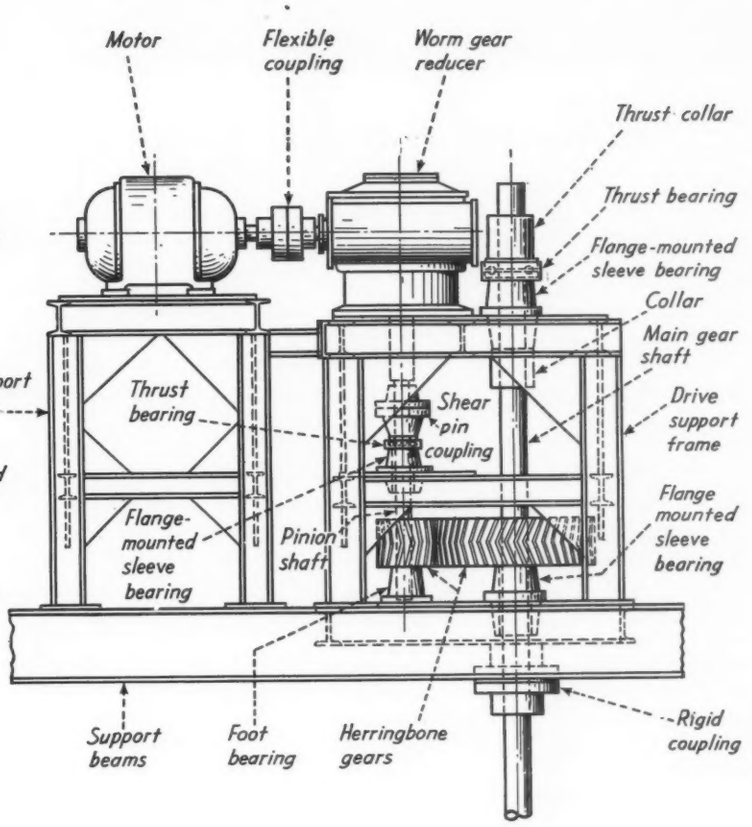


Fig. 6—Herringbone gears transmit power from pinion shaft, direct-connected to worm gear reducer, to main drive shaft in this unusual drive for a large, slow-speed, heavy-duty installation. Flange-mounted sleeve bearings are used throughout. Shear pin coupling on pinion shaft prevents damage to drive units.

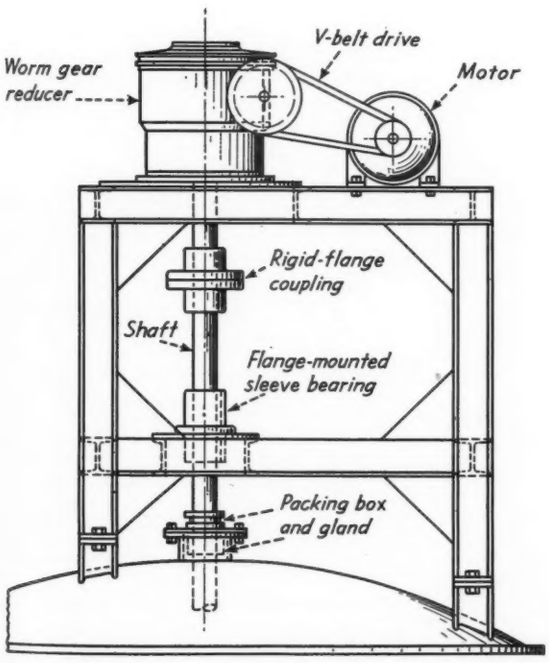


Fig. 7—V-belt drive from the motor to the worm gear reducer provides an additional speed reduction in this large heavy-duty unit. Otherwise this installation is similar to Fig. 4 at left. Great rigidity is provided by the heavy welded construction of the supporting frame.

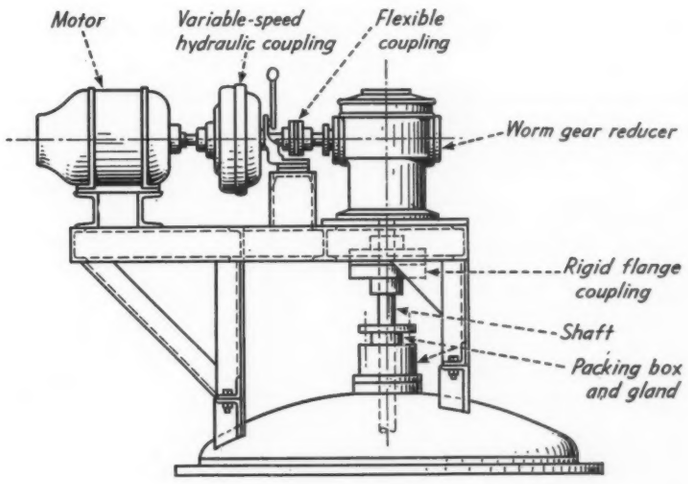


Fig. 8—Hydraulic variable speed coupling is used between the motor and the worm gear speed reducer of this light or medium-duty drive. Coupling gives speed control in addition to protection from overloads. No auxiliary bearing is used, the reducer bearings providing sufficient rigidity.

Question and Comment

Discussion of Helical Spring Tables

[*Editor's Note*—Following the appearance of the first part of "Helical Spring Tables," by Paul Klamp, in PRODUCT ENGINEERING for August 1939, Mr. Victor Tatarinoff, who himself has contributed spring design articles, initiated a discussion with Mr. Klamp on certain basic methods employed in developing the spring tables. The correspondence printed here presents some interesting points, both pro and con.]

Selection of Permissible Stress

MR. TATARINOFF: Special attention should be called to the selection of values for permissible stress in torsion given on page 362. Permissible stress may be established on the basis of physical properties of spring material (ultimate strength in torsion) and a reasonable factor of structural safety. Values of ultimate strength are supplied by material testing laboratories; factors of structural safety can be derived from practice, considering the nature of the duty for which the springs are required.

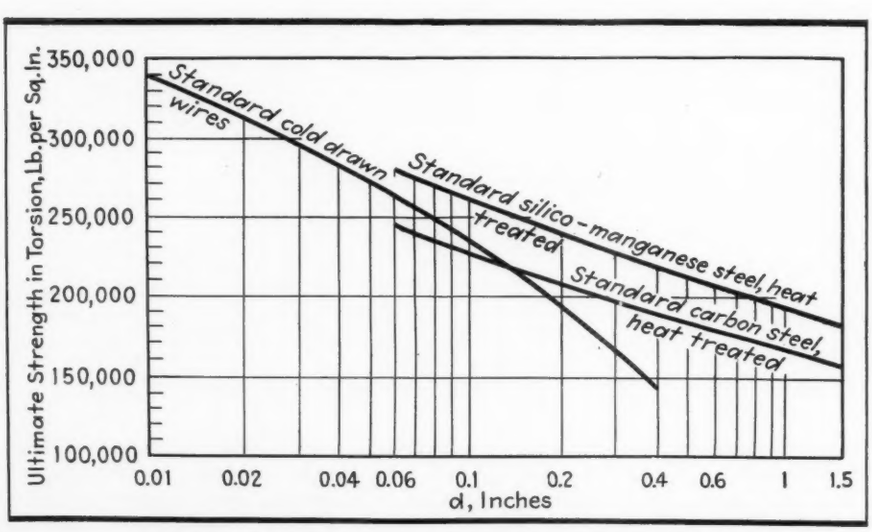
Mr. Klamp establishes three stress groups corresponding to certain sizes of spring material.

Using a certain average of stress will give confusion results. Utilizing the data for high stress group (81,000 lb. per sq.in.) 3½ to 29 per cent deviations are unavoidable. Yet more striking results will be discovered controlling the data of Table B on page 364. The actual permissible stress of 0.014 in. music wire is 325,000/3 = 108,300 lb. per sq. in. For largest size of spring material, 3/32 in., the stress will then be 236,000/3 = 78,700 lb. per sq. in. only, thus discovering a 37 per cent deviation. Ultimate strength, from the chart below, was divided by a safety factor of three in the above equations.

MR. KLAMP: Mr. Tatarinoff suggests that the load ratings and deflections should have been based on a different stress for each wire diameter listed. But, there are several reasons for employing the system of grouping the wire diameters and stepping down the stresses for each successive larger group.

In the first place, a table of this kind should cover various spring wire materials in order to be of practical value. Therefore, there is not much to be gained by using a different stress for each wire size, because this stress would represent a theoretical value for one kind of spring steel only.

In the second place, the stress on which the dimensions of a spring with rapid reciprocation are based is not a set value for a given wire size that can be derived by using a definite safety factor. Such a procedure would fail to take into account the relation between stress range and permissible stress.



Ultimate strength in torsion of common spring materials

In the third place, stepping down the stresses for groups of wire diameters corresponds to widespread practice in this country, as evidenced by "Standards for Spring Design" by W. M. Griffith, published in PRODUCT ENGINEERING, March and April 1937, and further by the spring tables published in data sheet form by *Machinery* in February 1938, compiled by J. I. Hommel of the Westinghouse Electric & Manufacturing Company. Many other publications recommend stresses without adjusting the stress to each wire diameter, including catalogs published by leading spring manufacturers.

Mr. Tatarinoff questions the use of the same stresses for wire diameters from 0.014-in. to 3/32 in. whereas, in the case of music wire, the ultimate torsional strength of 0.014-in. wire is 37 per cent higher than that of 3/32-in. wire. However, out of a total number of about 430 springs listed, there are but eight with a wire size of less than 0.023 in., for which in the case of music wire the safe stress could be up to 20 per cent higher than the maximum stress used in the table.

Incidentally, safe stresses and safety factors are, by no means, absolute values, as illustrated by the fact that, among nine leading automobile manufacturers in this country, one was using a maximum working stress as low as 60,000 lb. per sq. in., while another did not hesitate to go up to 83,000 lb. per sq. in., or 38 per cent higher. This difference between stresses for automobile valve springs is but one example of how spring users will not always accept one definite stress or safety factor, even for springs of similar nature.

Deflection Correction

MR. TATARINOFF: Mr. Klamp has omitted to introduce the deflection correction factor expressed by formula $y = 1 + 1/2c - 1/2c^2$, in which $c = 2r/d$. The actual deflection values will then be from 3 per cent ($c = 15$) to 10 per cent ($c = 4$) higher.

MR. KLAMP: Mr. Tatarinoff states that a correction factor for deflection was disregarded in Table B and cites a formula for this factor without giving its source. This formula actually would indicate errors as high as 10 per cent. But, such figures do not at all agree with facts. I like to refer to A. M. Wahl's work on "Factors Affecting Accurate

Spring Selection," published in *Machine Design*, August, 1939. Mr. Wahl has investigated the Goehner formula, covering the correction factor for deflection, published in Germany in 1930, and shows that it agrees very closely with tests he has conducted, as well as with the findings of other authorities. But according to this verified formula, the deflection error for any spring listed in the new table is well under 1 per cent, thus having no importance for general purpose springs where the torsional modulus alone may vary several per cent, even for the same kind of material.

Silico-Manganese Steel

MR. TATARINOFF: In studying the materials indicated in the Table A, I have not found standard silico-manganese heat-treated steel having widespread use for manufacturing springs. (Shock absorbers in railway cars, valve springs.)

MR. KLAMP: Silico-manganese heat-treated steel was omitted from Table A because sufficient data on endurance limits for this steel were not found. For valve springs, spring manufacturers usually furnish their own special steel.

Load Increase Factor

MR. TATARINOFF: I am obliged to point out that the stress group subdivision for certain materials on basis of load increase factor i should be properly explained, as no experienced spring designer will have the idea to reduce the permissible stress according to certain modification of spring preloading. It is difficult to understand why, for instance, the spring designer using carbon steel pretempered will be obliged to select 20 per cent lower values of permissible stress at increasing factor i from 1.5 to 1.667 (10 per cent only).

MR. KLAMP: The load increase factor i is, of course, a direct expression of the stress range on which the permissible stress in a rapidly reciprocating spring depends. The relation between stress range and stress is shown by numerous test results reported by F. P. Zimmerli in *Engineering Research Bulletin* No. 26, July 1934, published by the University of Michigan. Further test data on the effect of stress range on safe stress were reported by E. W. Stewart in PRODUCT ENGINEERING, August 1936 and by W. M. Griffith in PRODUCT ENGINEERING, March and April 1937.

Stress recommendations accompanying the new spring tables were based on these test results, with the limitation that there is not a stepless change from one stress to the next lower or higher one. However, it answers the purpose of enabling the designer of general type springs to take advantage of latest spring developments and yet greatly reduce the time required for spring selection.

Short-Cut Mathematics Solves Flagpole Problem

To the Editor:

The solution of Kenneth W. Winkle's flag pole problem, as set forth in the December PRODUCT ENGINEERING can be made much easier by use of a few mathematical time-savers.

Let the different sections of the pole be given the lengths a, b and c, as shown in the figure. Also let $10 = k$, and $100 = k^2$, for greater ease of handling the algebra. Then,

$$b^2 = a^2 + k^2 \qquad (1)$$

$$\frac{b}{a} = \frac{b+c}{a+k} \qquad (2)$$

$$a + b + c = k^2 \qquad (3)$$

Eliminating b and c and solving for a in terms of k:

$$b + c = k^2 - a$$

and

$$b + c = \frac{b\,(a+k)}{a}$$

Therefore,

$$b = \frac{a(k^2 - a)}{a + k}$$

Squaring this value of b and setting it equal to b^2 from Equation (1),

$$\frac{a^2\,(k^2 - a)}{(a+k)^2} = a^2 + k^2$$

Clearing this of fractions, multiplying out, collecting, and arranging the resulting terms in descending powers of a gives:

$$a\,(1+k)\,a^3 + k\,(2 - k^2)\,a^2 + 2k^2a + k^3 = 0$$

Now substitute 10 for its equal, k:

$$f\,(a) = 11\,a^3 - 490\,a^2 + 100\,a + 500 = 0$$

This is a rather mean cubic. It may be solved by several methods, but since practical, rather than theoretical, results are desired I recommend an approximation method.

We have a guide to the limiting values of a, the distance from the garage roof to the break in the pole. It cannot be greater than 45 ft., because if it were the top would not reach the ground. On the other hand, inspection of the figure suggests that $(b + c)$ is probably not more than about 20 ft. greater than a, for which figure a would be 40 ft.

The answer therefore probably lies between 40 and 45. Substituting

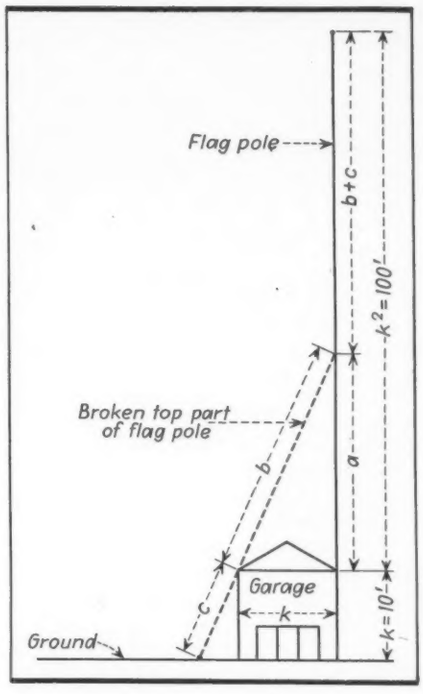
$$a = 40, f\,(a) = -75{,}500$$
$$a = 45, f\,(a) = 15{,}925$$

The hypothesis is right because the values are each side of zero, the theoretically true result.

A few more trials yielded $f\,(a) = 61$ for a value of $a = 44.32$ ft. This is probably close enough.

These substitutions, incidentally, can be carried out conveniently and rapidly by the method of synthetic division, a short demonstration of which follows.

First, set down along the top edge of



the sheet the coefficients of the polynominal in order of descending powers of the independent variable, with their signs. Then multiply the first coefficient by the value to be tried, set the product below the next coefficient and add. For example, to substitute $a = 40$, multiply $40 \times 11 = 440$. Put this below -490, and add. Multiply remainder again by 40 and add the product to the next coefficient, and carry out for all coefficients:

+11	−490	100	500
	+440	−2,000	−76,000
	−50	−1,900	−75,500

which remainder is the value of the polynominal $f(a)$ for $a = 40$.

Any desired degree of accuracy may be attained, of course, by successive substitutions. —LAURENCE F. SOUTHWICK
Meriden, Conn.

Inlay Actually Cuts Into Wall of Hard Plastic

To the Editor:

We read with considerable interest the article "Joining Metal and Plastics" by J. Earl Simonds in your November number, particularly with reference to our Chilton process. We are not completely in accord with Mr. Simonds' description of our process on page 495 of your November number.

He states that in the case of cellulose acetate parts the edges of the inlay cut into the plastic walls of the recess. He states in the next sentence that in urea and phenolic plastics, the inlay does not actually cut into the plastic material but causes it to be indented. The point to which we take issue is with reference to

the urea and phenolic substances.

We have inlaid hundreds of thousands of units molded of all types of materials. The top inlay does interlock with the walls of the recess. The walls are not under-cut but are perpendicular to the floor of the recess. If the hard materials were only indented by the inlays, the inlays would not stay permanently locked in the material—but they actually do. We have enough performance back of us now, so that we hardly see any grounds on which the facts may be disputed.

—James J. Larmour
Plastic Inlays, Incorporated

Seeks Design Data On Steam Jet Ejectors

To the Editor:

We would appreciate it very much if you can advise us where we can obtain design data in connection with the thermo-compression of steam in jet ejectors.

We have certain steam engines which exhaust into the dryers of paper machines, necessary make-up steam being added to the dryer steam header through a reducing valve.

We are considering the possibility of lowering the back-pressure on the engines, and thus increasing their available power, by feeding the make-up steam into the header through a steam jet exhauster which would serve the two-fold purpose of lowering the back pressure on the engine and increasing the pressure of the steam supplied to the dryers.

Apart from general theory on the flow of steam through nozzles, we have been unable to locate any design data giving such essential information as the most effective proportions to adopt for the jets, best velocities of flow, angles of diffusion, and the pros and cons of multiple or single jets, efficiencies obtained, etc.

We realize that the design of this equipment is to some extent a specialty but we have become so accustomed to look for, and to find, first class design data in Product Engineering that we confidently ask for your assistance in this problem.

—A Cunningham
Price Brothers & Co.

[*Editor's Note*—Probably the most comprehensive treatment of the theory behind thermo-compression will be found in the book "Steam and Gas Turbines," by Prof. A. Stodola. However, even a comprehensive theoretical treatment such as appears in Dr. Stodola's book or in any similar publication would be of little help in the actual design of such a unit. Experience is worth more than theory.

There are several companies which specialize in the design and manufacture of such equipment. The most practical procedure would probably be to get in touch with them, outlining the conditions of the problem, and getting from them their recommendations.

This is our answer. We know it's not a complete solution, but throw open the question to those of our readers who, because of their experience in this type of design, can offer a better answer.]

Air-Sealed Construction Of Stratoliner Cabin

To the Editor:

On page 537 of the December, 1939, number of Product Engineering the Boeing 307 Stratoliner for high altitude flying is briefly mentioned. The paragraph states that the internal pressure of the cabin is kept above that of the air outside for greater passenger comfort when flying at altitudes up to 20,000 ft.

I should think that the job of sealing the cabin would necessarily involve many interesting design features. For instance, all windows, doors, fittings where the controls pass through the cabin, etc., to say nothing of the fuselage itself, must be nearly perfectly air-proof to keep inside conditions really comfortable.

Can you give me further information concerning the methods employed to maintain the pressure?

—Stewart Dickinson
Detroit, Mich.

[*Editor's Note*—As Mr. Dickinson has correctly surmised, the cabin details of the Boeing 307 are indeed interesting. At altitudes of 13,000 ft., where the pressure is down around 8.6 lb. per sq. in., the atmospheric conditions begin to make most persons mentally drowsy and physically sluggish. At altitudes of 20,000 ft. the pressure not only drops to about 6.7 lb. per sq. in., but also the temperature drops to about 10 deg. below zero F. Therefore, in order to gain the advantages of flight at 20,000 ft., the cabin must be supercharged and heated.

The fuselage of the Stratoliner, constructed to withstand a pressure differential of 6 lb. per sq. in., is round in cross-section, this being the best shape for a streamlined pressure vessel. Windows are extra-heavy to withstand the pressure, in some designs being of double-pane construction with each pane set in rubber gaskets. Doors close against soft rubber gaskets, and are screwed down tight. We have recently heard of a design in which the door is closed against a rubber tube, the seal being completed by pumping a few pounds air-pressure into the tube after closing the door.

Fresh air, drawn through the leading edge of each wing, is compressed by two engine-driven superchargers and circulated throughout the cabin after being heated by means of freeze-proof condenser type steam radiators. Spent air is discharged through an exhaust chamber below deck. Pressure regulating apparatus automatically controls the superchargers to keep the pressure differential at 2½ lb. per sq. in., which is sufficient to create an apparent altitude of 12,200 ft.. when the plane is 20,000 ft. high.]

Can You Work This One?

H. E. SMITH

Solution to December problem—

Alphabet Division

If last month's problem was done the hard way, the letters would be replaced by numbers to give this result:

```
246)61998(252
    492
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However, if the reader tried to guess the indispensable link between design engineers and production men, he perhaps would have immediately arrived at "blueprints" which is the correct word formed by placing the letters in order of the numbers they substituted.

This Month's Problem—

Brain Testers

1. If a straight well could be bored through the center of the earth to China, and a pebble dropped from the exact center of the well at the surface of the earth, would the pebble strike the wall of the well, or would it pass unimpeded through the center of the earth?

2. A bottle and a cork cost $1.05. The bottle cost a dollar more than the cork. How much did the cork cost?

3. Find a whole number ending in 2 which can be exactly doubled by merely removing the 2 from the right hand end and making it the first digit.

4. If 3,001 digits are required in numbering the pages of a book, how many pages are there in it?

5. A fish weighs 10 lb. plus half his weight. How much does it weigh?

News and Summaries

1,400,000-Volt X-Ray Tube Built For National Bureau of Standards

Giant tube, to be used in extending high-voltage measurement work of Bureau, is constructed in ten sections

X-RAY DOSAGES greatly in excess of any now used by the medical profession can be produced by a new 1,400,000-volt X-ray tube now being built by the General Electric Company for the National Bureau of Standards. The complete equipment, including one of the most powerful constant-voltage, d.c. generators ever made, will be used for research work in the high-voltage field.

The tube, rated at 1,400,000 volts and 10 milliamp., is composed of ten sections of 140,000 volts each. It is mounted vertically and the envelope, consisting of ten glass cylinders placed end to end, will be mechanically shielded against breakage throughout its entire length.

For the satisfactory operation of the tube it is essential that all of the electrons leaving the cathode should reach the anode 24 ft. below—in other words, that none of the electrons should strike either the glass walls of the tube or the various metal accelerating electrodes. This necessitates not only starting the electrons in the right direction, but guarding them from the deflecting action of external electrostatic or magnetic fields. It also necessitates the attainment throughout the length of the tube of a high degree of symmetry.

Construction of the tube permits a unique method of applying energy to the electron beam after leaving the cathode. Starting at zero velocity, the beam is accelerated through the first section of the tube to 140,000 electron volts velocity. As it passes through each section of the tube it receives an additional voltage kick of 140,000 volts which gives it a final velocity of about 180,000 mi. per sec. and the energy it would have had it been given a single 1,400,000-volt kick. The tube can be operated at any lower voltage down to 300,000 volts.

The high-voltage generator furnishing power to the tube is composed of step-up transformers, capacitors, and kenotron rectifier tubes. It also has ten sections, each one connected to a section of the X-ray tube. The generator forms a stack 32 ft. high and 5 ft. in diameter. Corona shields of spun aluminum surmount the tops of the tube, the generator, and the resistance stack. Also, around each section of the tube and generator a doughnut-shaped corona shield is placed.

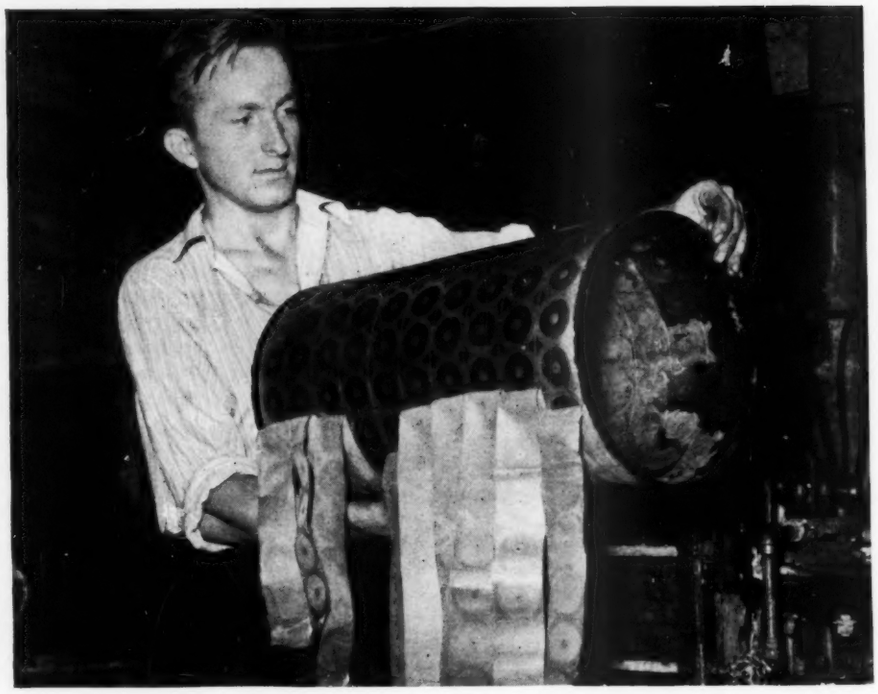
The new device will serve several important purposes at the Bureau. It will be employed to measure and set standards for X-ray dosages higher than those now medically available. The same high-voltage generator, with another multi-section tube, can later be used for the production of neutrons and artificial radioactivity. It can also be used to extend other high-voltage measurement work of the Bureau to higher levels.

Assembling doughnut-shaped corona shields on three top sections of giant X-ray tube

Vibrationless Engine Demonstrated in Sweden

A ONE-CYLINDER DIESEL engine designed with a radically new principle of vibration compensation has just been demonstrated in Sweden, according to a report to the Department of Commerce. The 10-hp. experimental engine, which successfully operates with practically no vibration, is regarded by Swedish experts as a most important step in the development of a completely vibrationless reciprocating engine.

The newly developed Swedish motor is a single-cylinder, theoretically compensated engine. The main part of the motor body serves as the compensating agent, having been transformed into a



Lektromesh metal screen, electroformed into smooth metal mesh, is continuously stripped from this cylinder matrix in rolls up to 36 in. wide and 1,400 ft. long. The new screen, produced in nickel, copper and other materials, can be readily fabricated by drawing, stamping, expanding, welding or soldering. Any desired design can be made in sizes from 25 to 400 mesh, with openings ranging between 15 and 50 per cent total area. The design is transferred to the cylinder matrix by a succession of sensitizing, etching, filling and surface treating steps, such that the cylinder will receive the electrodeposit uniformly only where desired. Developed by C. O. Jelliff Mfg. Corp. to supplement wire screen, Lektromesh has already been applied as fuel strainers and radio speaker screens.

mobile counter-balance for the piston movements. Both the reciprocating and the rotating parts and the power impulses from the one-cylinder engine are theoretically and practically given perfect compensation.

In contrast, the majority of efforts in the direction of vibrationless engines have tended toward construction of multi-cylindered machines in which the moving parts of the various cylinders compensate each other in the greatest possible degree.

Effect of Crystal Size In Bearings Discussed

OPTIMUM VALUES of rate of wear and equilibrium pressures are obtained in bearings which have a very small crystal size, states a paper presented by John R. Connelly at the Annual Meeting of the American Society of Mechanical Engineers. The results of investigations conducted on a high lead bearing metal indicate that, in the future, all bearings of this metal should be designed to make use of some type of insert that can be cooled rapidly upon pouring so as to produce small crystals. In corroboration of the results, it was pointed out that the automobile industry at present produces bearing inserts by a technique which gives very small crystals having a linear dimension in the order of 0.001 in.

The bearing metal selected for the investigation is one commonly used in industry. Its approximate composition is: lead 76 per cent, antimony 15 per cent, and tin 9 per cent. In the tests, both the chemical composition and the size of the specimen were kept constant. The crystalline structure was varied and the resulting changes in wearing properties were determined.

The test procedure consisted of pressing a machined flat surface of the specimen against a steel cylinder rotating in oil. A cylindrical groove was worn in the alloy, and as the contact changed from a line to a progressively larger area, measurements were taken to determine the volume of material removed per length of travel of the cylinder's surface.

Of possibly greater importance than the specific results on the effect of crystal size are certain implications in connection with fundamental analysis of a journal-lubricant-bearing combination. The present concepts of thin-film lubrication take no account either of the material or crystalline structure of the mating surfaces. A theoretical explanation of wear should wait on further research in the field of thin-film lubrication, and the explanation should be developed independent of the data in the range of thick-film lubrication.

Cross-Sectional Areas Of Structural Members

ERRORS which may creep into the measurements of cross-sectional areas of modern lightweight structural members call for a number of unusual precautions, a recent paper published in the *Journal of Research* reveals.

In introducing new types of construction, involving in many cases new materials and new methods of fabrication, the design engineer is guided by the stresses obtained from laboratory tests. These stresses, if they are based on the nominal cross-sectional areas, may be subject to an uncertainty of several per cent. For example, a thickness tolerance of 0.002 in. amounts to 5 per cent for sheet metal 0.04 in. thick. An accurate value of cross-sectional area will reduce this uncertainty and make possible more efficient design.

The cross-sectional areas of many of the lightweight members used in aircraft design cannot be obtained accurately from measurements of the cross-sections, because of their size or shape. In such cases, the average area can be obtained by a volumetric method wherein the volume of the member is divided by the length. The volume can be obtained by dividing the weight of the member by the density of the material, or by immersing it in a liquid.

If the area varies from section to section, the area of a given cross-section can sometimes be obtained from the average area of a short specimen taken from that location.

New Process Produces Stainless Clad Sheets

A NEW COMPOSITE METAL was recently introduced into the market now occupied by tin plate and galvanized sheet and plate, with the announcement of the Kinkead process for alloying the surfaces of carbon steel sheets and shapes with coatings of stainless steel or other metals.

Briefly, the process consists of fusing the alloying metals into the surface of an ingot or slab of carbon steel in an arc welding machine developed by R. E. Kinkead and manufactured by the United Engineering and Foundry Company. As an example, in producing a 17 per cent

chromium surface, a predetermined amount of powdered ferro-chrome is sprinkled on a slab. Furnace slag is spread over that to prevent oxidation, and heat from a carbon arc fuses the metals. A special acid bath removes scale before the slab goes for final rolling.

Alloyed coatings of molybdenum, nickel, titanium, or silver can be produced in the same way, in practically any thickness desired up to 10 per cent total thickness. Principal uses foreseen for the new composite metal are in railroad rails, which must be tough and hard at the same time; all types of kitchen and laundry equipment; air ducts; roofing; and many other similar products.

Since no large scale production has been started, it is too early to predict just how the material can be adapted to high-speed fabrication methods. Research to date shows little difficulty in deep drawing, cold reduction, welding, soldering, pickling or polishing.

Patent Practices To Be Investigated

CERTAIN "economically unsound" applications of patent rights, claimed to be evasions of Federal antitrust laws, will be attacked by the Department of Justice in a program announced by a recent report. Attorney General Frank Murphy has authorized: (1) a general investigation, conducted primarily through a grand jury to be convened in New York, of the use of patents, and patent agreements by which certain industries in the United States are controlled and dominated by one or several large companies; (2) an equity suit against 12 corporations engaged in the business of producing glassware machinery and glass containers.

It is claimed that facts now in the possession of the Department indicate that patent rights have been used in general to restrain trade and eliminate competition. By reason of these practices, arbitrary and artificially high and rigid prices have been maintained in certain industries, the report states.

Congress is given the power to grant the patent privilege "to promote the progress of science and useful arts" and legislation authorizing the issuance of patents is the mere instrument through which the constitutional purpose is reached. The Department believes that the abuses of patent rights frustrate this constitutional purpose, and run counter to the policy of the antitrust laws.

The complaint in the equity suit charges that 12 glass corporations have, in violation of the Sherman Act and the Clayton Act, combined and conspired to restrain trade and secure a monopoly in machinery for the production of glassware in patents covering such machinery and that as a consequence competition has been stifled and the price of glass products artificially increased.

If the facts charged in this case are established as a violation of the antitrust laws, it is expected by the Department of Justice that there will be a restoration of competition in the glass machinery and container business. This field, states the report, will then be opened for the employment of additional capital and labor; prices and production will be established on a competitive basis; the stimulating effects of competition should be felt in the field of research and technological advancement.

Further details of the present patent situation will be given in articles by Karl Fenning, formerly Assistant Commissioner of Patents and now Editor of *U. S. Patents Quarterly*.

Meetings

SOCIETY OF AUTOMOTIVE ENGINEERS—Annual Meeting and Engineering Display, Book-Cadillac Hotel, Detroit, Mich., January 15–19. John A. C. Warner, secretary, 29 West 39th St., New York, N. Y.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS—Winter Convention, New York, N. Y., January 22–26. H. H. Henline, secretary, 29 West 39th St., New York, N. Y.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—Sixth International Heating and Ventilating Exposition, Lakeside Hall, Cleveland, Ohio, January 22–26. Charles F. Roth, Grand Central Palace, New York, N. Y.

NATIONAL SOCIETY OF PROFESSIONAL ENGINEERS—Fifth Annual Convention, Netherland Plaza Hotel, Cincinnati, Ohio, January 25–27. Edward Larson, chairman, 7 W. 6th St., Cincinnati, Ohio.

Do You Know That—

PHOTOELECTRIC CONTROL TUBES have been applied to four-color printing processes to keep the color register within tolerance limits at operating speeds of 1,000 ft. per min. (1)

GERM-KILLING ultra-violet lamps are being tried out in air-conditioning systems. Investigations show that one 15-watt lamp can sterilize 200 cu. ft. per min. of air in a duct of non-reflecting material. (2)

A THIN FILM OF GOLD as the foundation upon which is deposited the fluorescent material will improve the brightness of television images. The gold film has the same color as the light emitted by the willemite, and attracts the electrons so that they strike the screen with greater force. (3)

Electrons entering a salt crystal form the heart-shaped cloud effect in this image being projected by Westinghouse's Dr. L. J. Berberich. The crystal has been heated to a temperature of 1,200 deg. F. in a special electric furnace. Electrons reveal their presence as a reddish mass, as their vibrations absorb all but the red and violet colors of the white light passed through the crystal by the projector.

New Materials and Parts

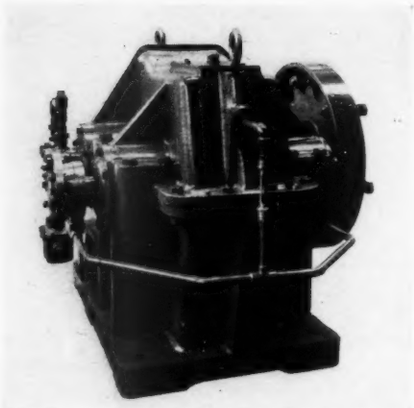
Sealed Micro Switches

Synthetic rubber seals and gaskets on these precision milli-break switches give positive protection against dust, moisture or oil which would impede the free movement of the plunger under light pressures. On the bottom of the switch where it is open for electrical connections, a flat gasket of synthetic rubber is used. Screw holes are sealed by lead washers. A flexible synthetic rubber gasket shaped like a bellows, fastened to

the top of the plunger and to the housing of the switch, allows the plunger to operate freely in the switch housing. The special shape and composition of the bellows gives a flexibility which does not affect the sensitivity of the switch. Micro Switch Corp., Freeport, Ill.

Speed Increasers

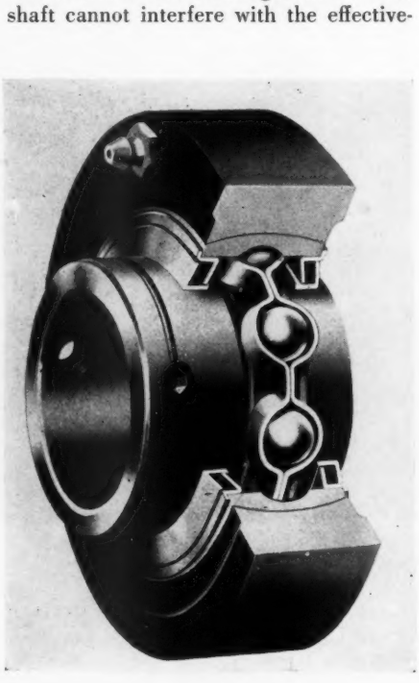
New line of speed increasers is designed for operating centrifugal pumps, high-speed blowers, compressors, and for pipe-line service. Speed increasers can be supplied for either right- or left-hand assembly. Sleeve bearings have been designed to provide low unit pressures and to assure permanent alignment with minimum friction loss. Lubrication is accomplished by a self-contained circulating oil system. Gears are spray-lubricated at the line of contact; all bearings are under pressure lubrication. Frames are split horizontally to permit easy access to all internal parts, and each unit has an easily removable inspection plate. Average efficiency of these units is better than 96 per cent. Standard gear ratios vary from 2 to 1



to 12 to 1, and these new units are available in ratings of from 1 hp. per 1,000 r.p.m. of the high-speed shaft to more than 1,800 hp. per 1,000 r.p.m. Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

Bearing Cartridge Units

Featuring the same advanced design as other Sealmaster units, these new self-aligning ball bearing cartridge units have a centrifugal labyrinth seal which effectively seals the bearing from all foreign materials in the atmosphere and retains lubricant. Misalignment of the shaft cannot interfere with the effectiveness of the seal, for the outer race is ground on radius and locked in ground housing socket. Locking nipple permits 2 to 4 deg. misalignment in any direction. Wear or glazing of felts is positively avoided as they are assembled without pressure. Stephens-Adamson Mfg. Co., Aurora, Ill.



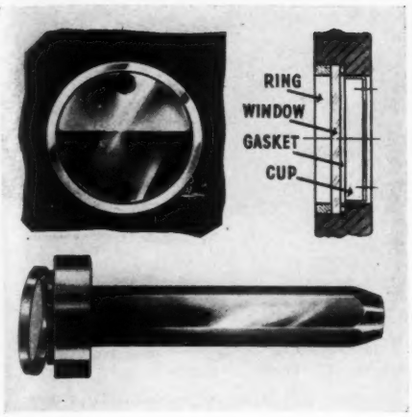
Stay-On Safety Nut

Designed and manufactured for industries whose fastening problems are made more difficult because of shock and vibration, this nut has a positive and automatic locking action which does not depend upon special threads, springs, pins or other extra parts. The locking portion of the nut consists of a raised crown section, slotted perpendicular to

the axis of the nut. This section is distorted so as to form an elliptical ring section. The nut is double heat-treated to give the crown a permanent set and tough resilient spring characteristic upon which the locking action depends. Gripping action of the crown on the bolt produces friction necessary to prevent the nut moving on the bolt. Nut can be applied repeatedly without loss of locking action and without harming the bolt threads. Lamson & Sessions Co., 1971 W. 85th St., Cleveland, Ohio.

Sight Oil Window

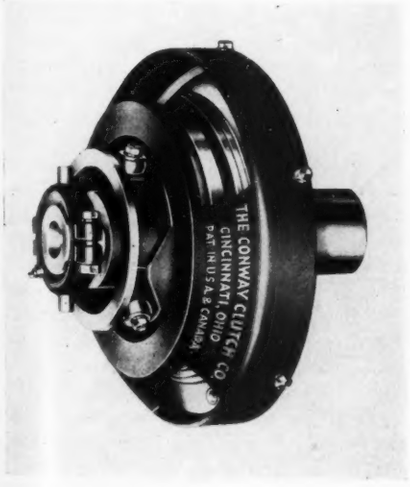
Improved visibility is afforded by a new type oil window for general use in connection with oil reservoirs, containers and lubricators. The assembly consists of chromium-plated ring with pol-



ished face pressed flush, window, gasket, and cup, all fitted snugly into a metal pocket. Drilled openings at top and bottom of cup permit free passage of the oil. Bright aluminum background of the cup acts as a reflector to give sharp visibility of the oil level. The self-contained unit is conveniently installed in a simple bored opening readily provided in the design of the machine casting or part. Standard sizes are ¾ in., 1⅛ in., 1⅜ in., specified according to diameter of reflector. Special assembly tool is provided. Bijur Lubricating Corp., Long Island City, New York.

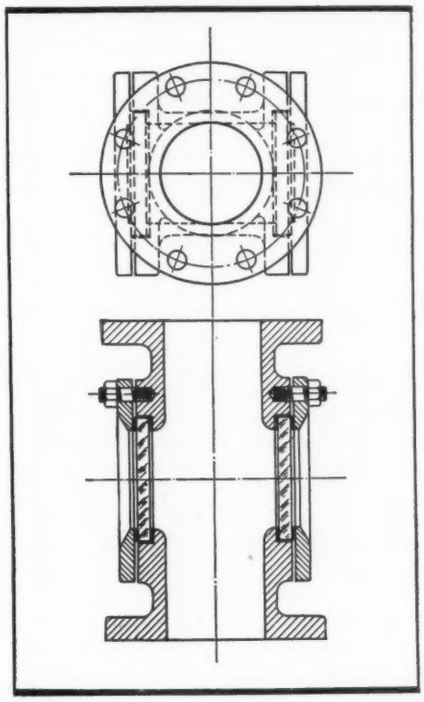
Automatic Release Clutch

Overload release for protection to mechanical parts is a feature of this new clutch, which can be set to transmit any torque from zero to its rated capacity. The clutch is engaged in the regular manner of disk clutches and smoothly transmits the load until an obstruction or excessive torque is encountered. When subject to overload, the clutch immediately slips. After slipping for one-half a revolution, it automatically disengages itself. If so desired, it can also operate a limit switch and disconnect electrical equipment. Adaptable to a variety of machines and conveying systems where an obstruction can cause costly damage to some machines. The Conway Clutch Co., 1543 Queen City Ave., Cincinnati, Ohio.



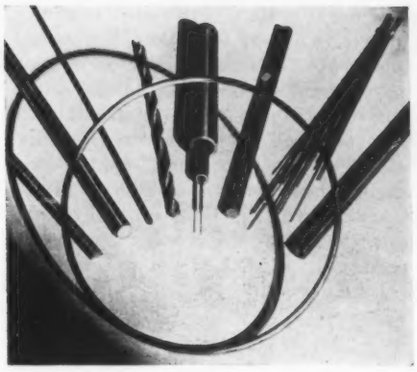
Visible Flow Fitting

Designed for any purpose where visual inspection of the flow of fluids under pressure is desired. The body of this visible flow fitting is of cast iron and the window-frame is of steel. Windows are made of Pyrex glass, and are 4¼ in. in diameter for all sizes. Fitting is flanged for installation in the line and all sizes measure 13¼ in. between finished surfaces. Available in six sizes for 2½, 3, 4, 5, 6, and 8 in. lines. Cochrane Corp., 17th St. and Allegheny Ave., Philadelphia, Pa.



Extrusion Process For Cemented Carbide

Carboloy cemented carbide can now be produced in the form of tubing, spirals, and round or shaped bars by means of an extrusion process. The rods, spirals and tubes are available in lengths up to 20 in. and within a diameter range of 0.015 in. to ⅜ in. The tubing can be made as small as 0.06 in. outside diameter by 0.03 in. inside diameter, leaving a wall thickness of 0.015 in. By means of a supplementary process, the cemented carbide rods and tubing can be bent to various shapes. With the new processes, parts can be formed directly into desired shapes, eliminating most of the customary hand-forming operations. Carboloy Co., Inc., 11185 E. 8 Mile Road, Detroit, Mich.

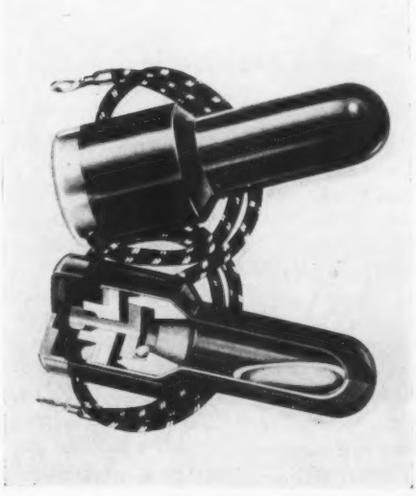


Porous Bronze Retainers Lubricate Ball Bearings

For specialized applications, new ball bearings incorporate retainers made of oil-impregnated die-pressed bronze. Porous structure of the retainer forms a reservoir for the retention of as much as 25 per cent of its volume of oil. The oil is fed from the retainer at the correct speed for proper lubrication, but not sufficiently fast to cause leakage. In sub-zero applications, where even light grease will harden and cause extremely high starting torque, these bearings provide a minimum of drag. Fafnir Bearing Co., New Britain, Conn.

Unbreakable Mercury Switch

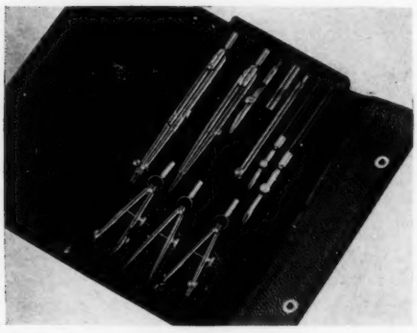
Recently improved Durakool metal mercury switch features double reduction, which prevents deterioration of the mercury, eliminates possibility of con-



tact interruptions, increases efficiency and reliability of operation. Silent, unbreakable, non-inflammable, Durakools are available in sizes up to 200 amp. capacity and operate on small tilt with little energy. The improved mercury switch allows many new applications ranging from wall switches to motor starters, and is now in use on electrical devices requiring from a few operations per month to 2,400 per min. Durakool, Inc., Elkhart, Ind.

Drawing Instruments Made in United States

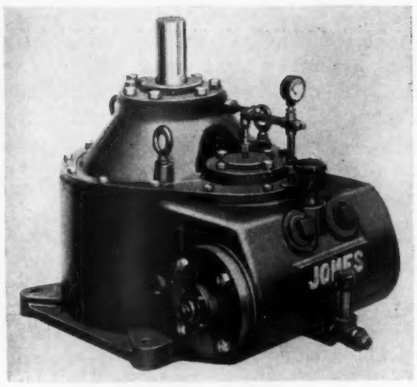
American-made drawing instruments, called Minusa instruments, are developed for straight line production. They have a slender rounded taper construction designed to produce excellent balance and a maximum of strength. Use



of a special hard rolled nickel silver stock lends unusual rigidity to compasses, dividers and springbows. Careful hand fitting of all moving parts insures smooth, even action. Except for the hand fitting operations, Minusa instruments are made by modern machine production methods throughout, actually improving on handmade instruments imported from Europe. Available in complete 10-piece sets. Keuffel & Esser Co., Adams & Third Sts., Hoboken, N. J.

Worm-Helical Speed Reducers

These new worm-helical speed reducers are designed for continuous operation coupled directly to electric motors. Steel worms are case-hardened and ground. Worm gears are of chill cast nickel alloy bronze. Low-speed helical gears are of manganese alloy cast steel, heat-treated. Each unit is supplied with an oil pump mounted on the rear worm bearing cover plate, and driven directly from the worm. Unit can be furnished in many mounting styles and in various standard assemblies. Also available are different styles of bearing mountings for



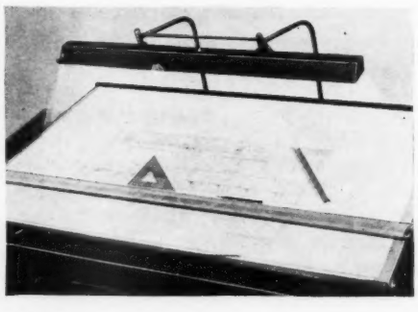
the vertical low-speed shaft. Ten sizes of standard worm-helical reducers are available, each with 15 standard speed ratios ranging from 40 to 1 minimum to approximately 250 to 1 as the maximum. Resulting range of capacities covers practically the entire industrial field. W. A. Jones Foundry & Machine Co., 4401 Roosevelt Rd., Chicago, Ill.

Aluminum Alloy Bearings

A new aluminum alloy automotive bearing has outstanding advantages of high fatigue strength which is many times that of conventional bimetallic bearings used for the same purpose; high resistance to corrosion; and economical construction. In tests recently made, the all-aluminum bearings were shown not only to have a superior resistance to fatigue, but also were found to be satisfactory as far as friction is concerned. The aluminum bearing is not difficult to fabricate and can be finished by customary methods. Use is at present limited to engines for commercial vehicles which have hardened shafts and which operate at relatively lower speeds at the bearing surfaces. Aluminum Co. of America, Pittsburgh, Pa.

Drafting Light

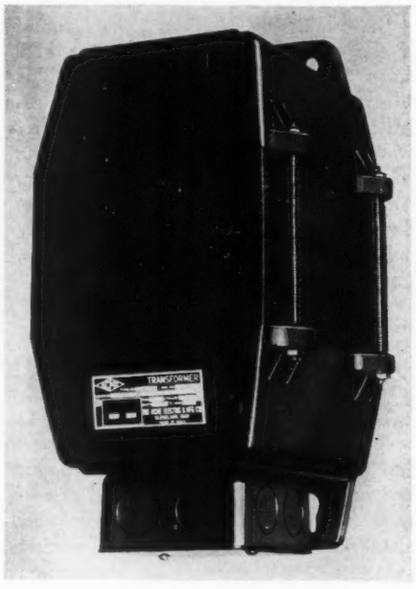
This new No. 1810 drafting light is of the fluorescent cold daylight type. It provides the required light intensity, makes color matching easier, and is cooler and easier on the eyes of the



draftsman. Special 44-in. reflector gives full range lighting over the drawing board, and eliminates the eyestrain which results from the contrast between light and shade. It can be set and locked at any angle, or it can be pushed back out of the way. The unit is finished in black crystalline enamel. Hamilton Mfg. Co., Two Rivers, Wis.

Transformers

High full load operating efficiency is maintained in this entirely new line of air-cooled transformers through design and construction features such as full air draft external ventilation, specially wound hand finished coils insulated between layers with high dielectric strength



insulating materials, cores of thin gage low loss steel. The balanced proportion of core and coil, and general design materially increase the operating efficiency. Three separate streamlined models cover all standard voltage characteristics from 3/4 to 10 kva. Acme Electric & Mfg. Co., Cleveland, Ohio.

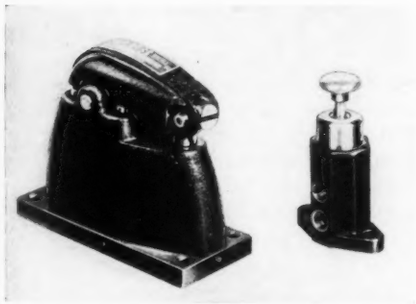
Geared Motors

Type KD is a heavy-duty, double-reduction geared motor with output at armature shaft of approximately 1/10 hp. at 7,000 r.p.m. Gear unit bearings are of composition bronze lubricated by light grease bath. Housing is of cast iron, finished in black wrinkle enamel. Motor is mounted by means of a machined face, a pilot boss and four tapped holes around the output shaft. Gearing is of the worm type, with hardened steel worms functioning with a bakelite gear in the first stage and a hardened

steel gear in the second stage. Available in six standard ratings with ratios ranging from 99:1 to 1,024:1, for either a.c. or d.c. Dumore Co., Racine, Wis.

Pilot Controlled Valve

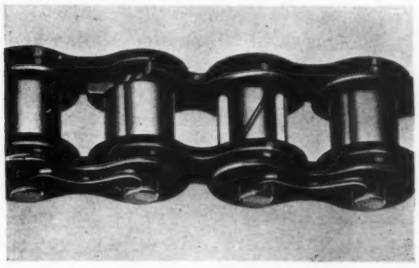
Designed for semi-automatic control, full automatic control, or remote control of double acting air cylinders, this new air-operated valve is actuated by pilot controls. These pilots are made in various designs to meet immediate control demands for hand, foot, or mechanical action with a push, cam, toggle or other desired movement. Although this new air-actuated valve is different in design, it retains the same fundamental poppet construction features, with its air seal and rapid action advantages, that are



used by the entire line of Ross air valves. The new system is especially well suited to manufacturing problems calling for a series of successive operations demanding rapid and accurately timed progressive movements. Ross Operating Valve Co., 6490 Epworth Blvd., Detroit, Mich.

Roller Chain

Channel-lubricated roller chain is a design which retains the self lubricating features of Morse roller chain and is completely interchangeable with all other standard roller chain. Oil applied on the outside surface of the rollers of any roller chain will seep under rollers and lubricate inner roller surfaces turning on the bushings. The difficulty in getting oil past the bushing to the core of the joint is overcome in this chain by diagonal channels rolled into the stock which extend through the side plates of the bushing link. The channels become in effect small pumps which literally force oil to all wearing parts of the chain. Rollers are hardened and



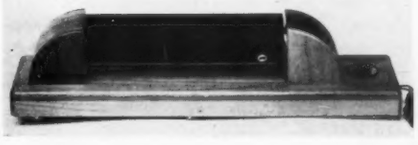
ground, turned out of solid bar stock alloy steel to attain greater accuracy and wearing strength. The cross section of the side plates has been increased. Morse Chain Co., Ithaca, N. Y.

Enamels for Plastics

Kem Plastite enamel is especially formulated for finishing plastic materials. Tests on a wide variety of plastic formulations indicate that this finish withstands severe abrasion and impact without marring or chipping. Another finish of similar properties, known as Kem Bakolescent enamel, gives a beautiful rich opalescent finish which can be baked without occurrence of unsightly flooding. Advantages claimed for the use of color finishes on brown plastics in place of using color plastics are threefold: First, there is a very considerable saving in spray applying colors to the cheaper brown plastic material. Second, the appearance is generally superior, and richer colors are possible. Third, manufacturers are enabled to produce all plastic objects in the one cheaper compound and spray on whatever color is in current demand. Sherwin-Williams Co., Cleveland, Ohio.

Small Blueprint Machine

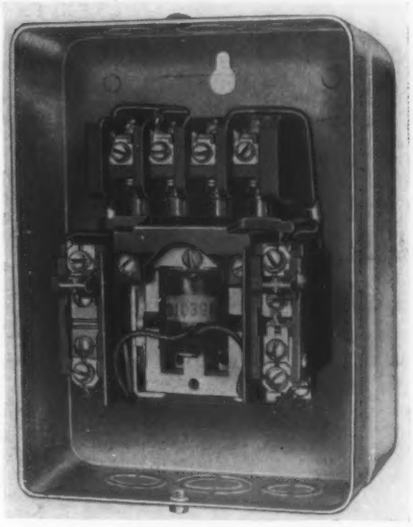
Baby Blue desk unit is only 34 in. long and 8 in. wide, and weighs but 15 lb. The unit prints black and white



prints as well as blue prints. Printing capacity is 10x18 in. Source of light is a tubular lamp which insures even distribution of light. The unit, finished in natural polished gumwood, is equipped with a time-switch so that the operator can perform other tasks while the print is being exposed. Vandenwood Co., 2900 Euclid Ave., Cleveland, Ohio.

Magnetic Starter

Type M-2 magnetic starter is particularly suitable for the machine tool and allied industries which require that starters for motors up to 2 hp. provide under-voltage protection and be electrically interlocked with other starters. Consists essentially of a 3- or 4-pole contactor with two hand or automatic reset isothermic temperature overload relays mounted on the contactor frame.



Contacts are double-break, silver-tip. All parts including contacts and coils are easily removable. Base is of non-hygroscopic molded material. Trumbull Electric Mfg. Co., Plainville, Conn.

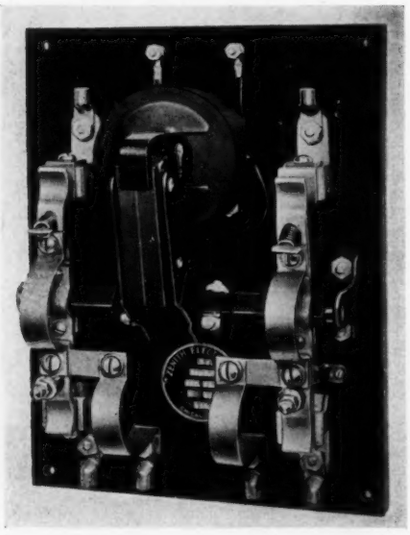
Black Plating Process

Described as producing a lustrous, deep black electrodeposit superior in color to any before known, this new process is based on a combination of molybdenum and nickel, available in plating salt form. Deposits are sufficiently black to escape color refraction under an intense beam of light in a dark room. Known as "moly-black," the process operates at a high rate of deposition at low current densities, twenty times more rapidly than nickel under identical conditions. Its throwing power is higher than any other plating system, giving deposits of uniform thickness even in deep irreg-

ular recesses. Applicable to bases of zinc; aluminum; cadmium electroplated on brass, iron or steel; tin; nickel; and steel or iron. Coating works well over any of the suggested base metals, particularly as a decorative finish for indoor use. For outdoor use, the black plate should be deposited over an undercoating of cadmium, after which it should be lacquered. Deposits are of low density, with a maximum thickness of 0.002 in. E. I. du Pont de Nemours & Co., Wilmington, Del.

Automatic Transfer Switch

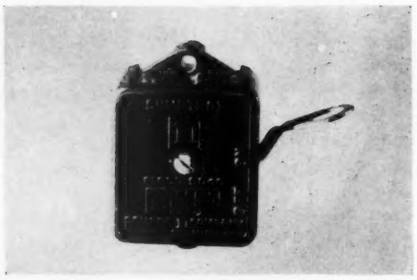
For automatically connecting a lighting or power load to an emergency source in case of failure, this automatic transfer switch is of the spring gravity drop-out type. Contacts are copper to



copper, and they operate with a rolling and wiping action. Compression springs provide required contact pressure and permit quick opening. The entire switch is compact in construction, and any part can be inspected or replaced without disturbing other parts in the assembly. Can be mounted in any position. The switch will transfer when the voltage of the circuit drops to 70 per cent, or less, and will restore the load to the normal source when the voltage reaches 90 per cent. The unit is available in current capacities from 30 to 300 amp. for all standard voltages. Mounted on ebony asbestos panel, with or without cabinet. Back-connected switches can be supplied. Zenith Electric Co., 845 S. Wabash Ave., Chicago, Ill.

Float Switch

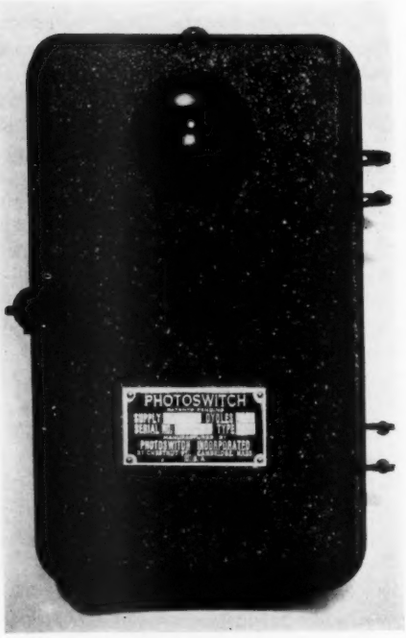
New Bakelite-inclosed, lever-operated float switch is designed particularly for use with sump pumps or cellar drain-



ers. A positive wiping action on the contacts is provided to eliminate pitting and burning. Wiring is accomplished by fastening four leads to three terminals, one terminal providing a common connection. Known as the Sumptrol, it is conveniently mounted by means of two screws through the integral flange. Electrical rating is 1 hp., 110 or 220 volts a.c.; ½ hp., 115 or 230 volts d.c. Square D Co., Detroit, Mich.

Photoelectric Limit Switch

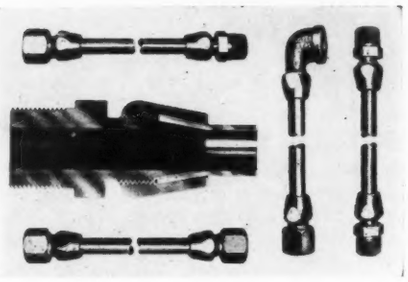
This new low-cost control is a simplified photoelectric unit, ruggedly constructed, suitable for installation in heavy industries where exacting conditions of vibration, humidity and high temperature are encountered. Type A14B limit



switch is supplied with light source and weatherproof housing suitable for installation in any location. Type A20C limit switch operates directly from the light given off by red-hot metal of any shape. The switch controls and limits the motions of heavy objects, such as billets and sheet steel, cranes and hoists, and red hot rods. Photoswitch Inc., 21 Chestnut St., Cambridge, Mass.

Connectors

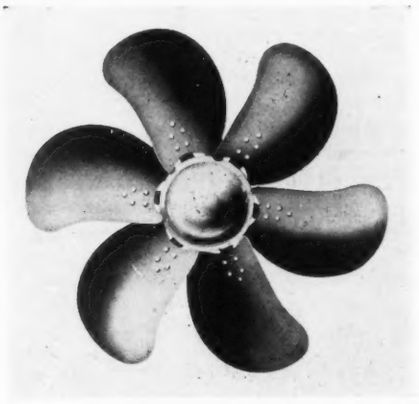
These connectors are available for use with aluminum, brass, copper and steel tubing, in sizes ranging from ⅛ in. to 2 in., for various adaptations. This type of connector employs a compression joint which is sealed both inside and outside with the ends of the tubing



flared to a 20-deg. angle. This provides a long flare which makes certain an absolutely tight seal with the tubing wedged between the self-aligning compression nut and the fitting. Leaks or breaks caused by the fitting cutting into the flared part of the tubing are impossible. Fittings can be loosened and tightened any number of times without injury to the tubing. Superseal Corp., 300 Fourth Ave., New York, N. Y.

Fan Assembly

Rigidity of the blades in this sturdy six-blade fan assembly is achieved by the use of hardened steel, giving the necessary strength and weight for extra-heavy duty. The blades are perfectly balanced by weight before they are assembled, requiring no drill-outs, no slugs nor any filling to make the assembly balance. The blades are heavy enough in construction to be used either



for direct or belt-drive application. The assembly will be manufactured in sizes from 30 in. to 72 in. William J. Lohman, Inc., 62 Ninth Ave., New York, N. Y.

Books and Bulletins

Problems in Mechanics

G. B. KARELITZ, J. ORMONDROYD AND J. M. GARRELTS—*271 pages, 6 x 9 in. Clothboard covers. Published by Macmillan Co., 60 Fifth Ave., New York, N. Y. Price $2.50.*

This volume contains a collection of problems covering statics, kinematics and dynamics, which are based on the original collection of the late I. V. Metschersky, of the Polytechnic Institute of St. Petersburg. The volume begins with a section of 50 pages giving a brief outline of the principles and theorems used in solving the 782 problems which follow the resume.

In the section devoted to plane statics are problems on trusses and cables. Problems on the first and second moment of areas are included in the part covering centers of gravity. Problems covering the kinematics of a point and of a rigid body in rotation about a fixed axis and motion parallel to a fixed plane are followed by problems in relative motion of a point and in composition of rotations of a rigid body.

Other problems deal with the application of Newton's differential equation to the motion of particles and to rotation and plane motion of rigid bodies. Special sections on bearing reactions, vibration, oscillation and impact are included.

Mills' Materials of Construction

EDITED BY LLOYD F. RADER—*Fifth Edition. 564 pages, 6¼ x 9¼ in. Maroon clothboard covers. Published by John Wiley & Sons, Inc., 440 Fourth Ave., New York, N. Y. Price $4.*

Now in its fifth revision, this book is useful to the designer who desires a concise, general text covering manufacture, properties and applications of engineering materials. The original 1915 edition was an outgrowth of Prof. Mills' engineering courses at Cornell and was revised in its second, third and fourth edition by Prof. Hayward of the Massachusetts Institute of Technology.

In bringing the book up-to-date, new chapters on organic plastics and organic protective coatings, glass brick, alloy cast irons, and a more extensive treatment of metallography have been included. Revisions have been made in the chapters on wrought iron, malleabilizing process, welding, copper ore extraction, properties and uses of non-ferrous metals including the die-casting alloys, and alloy steels.

The book is divided into eight sections: definitions; metals; cementing materials; concrete and stone; heat insulating and refractory materials; and organic materials. Copious references are furnished with each chapter. The value of the book is further enhanced by the inclusion of considerable tabular and graphical matter.

Public Speaking for Technical Men

S. MARION TUCKER—*397 pages, 5¾ x 8¼ in. Blue clothboard covers. Published by McGraw-Hill Book Co., 330 West 42nd St., New York, N. Y. Price $3.*

Aimed at the scientist and engineer, this book takes the point of view that a speaker's chief business is to hold the attention of his audience. This purpose excludes all "exercises" and specific instruction in speaking which can be found in many good manuals on the subject.

In achieving his purpose, the author attacks destructive faults common to technical men who mount the speaker's platform. He breaks away from the didactic, formal style of most textbooks by creating imaginary characters whose failures in speechmaking are strikingly similar to those of the average technical man. The free-running style of the book makes it easy for the reader to observe his own faults and to improve his method of presentation by accepting the author's helpful suggestions.

The topical order of the book is logical, beginning with the more obvious faults and working up to the less common and less perceptible. The two final chapters sum up the entire book and show its main ideas actually at work.

Graphic Presentation

WILLARD C. BRINTON — *620 illustrations, 512 pages, 6¼x9¼ in. Maroon flexible covers. Published by Brinton Associates, 599 Eleventh Ave., New York, N. Y. Price $5.*

This book is a dictionary and handbook of value to engineers and executives who wish to use graphic presentation in the preparation of reports. Its pages, printed in black and four colors, contain many illustrations of outstanding examples classified by type and use. Text is short, most of the points being clearly presented by the illustrations.

Many chapters of the book give a detailed description of all types of charts, graphs, tables, posters, maps, displays, dioramas, and photographic methods. Of specific interest to chief draftsmen, the book includes a discussion of methods for preparing graphic presentations, covering color and its use, reproduction methods, selection of paper, and listing drafting equipment especially adaptable to the work.

The book is cross indexed, and provides a new type of topical index for easy reference.

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Errors in Reflectance Measurements of Porcelain Enamel Surfaces

M. J. BAHNSEN and L. S. O'BANNON—*16 charts, 30 pages, 6x9 in. Published by Ferro Enamel Corp., Cleveland, Ohio.*

This bulletin reports an original investigation to determine the cause and nature of errors which exist in reflectance measurements of porcelain enamel surfaces, and to determine what precautions should be maintained to obtain maximum accuracy. The tests show that an accuracy of plus or minus 0.3 per cent is possible when large, carefully weighed, essentially flat samples are sprayed with minimum of orange peel and fired to correct temperature.

Lead Bronze Bearings

32 illustrations, 48 pages, 6x8½ in. Published by Copper Development Association, Thames House, Millbank, London, S.W. 1.

This well-illustrated booklet is an excellent general treatise on lead bronze bearings. Beginning with a discussion of the theory of bearings, it covers the properties and structure of lead bronze alloys. Developments in melting and casting techniques, design, machining and erection of the bearings, brought about during the past few years by the growth of the light engine industry, are set forth. The material was originally published by the Deutches Kupfer-Institut E.V., Berlin, as "Bleibronzen als Lagerwerkstoffe."

A.S.T.M. Standards on Petroleum Products

336 pages, 6x9 in. Heavy paper covers. Published by the American Society for Testing Materials, 260 S. Broad St., Philadelphia, Pa. Price $2.

This 1939 compilation of the A.S.T.M. Standards on Petroleum Products and Lubricants contains 62 standardized methods of test, ten specifications and two lists of definitions and terms. Several new standards are published for the first time covering carbon residue (Ramsbottom method), gum stability of gasoline, determination of tetra-ethyl lead in gasoline, consolidated specifications covering A.S.T.M. thermometers and others. Also included in the volume is the annual report of Committee D-2 on gum and tetra-ethyl lead. For information only, a proposed method of test for neutralization numbers of used aviation oils is published.

Manufacturers' Publications

Materials

BEARING ALLOYS—Lumen Bearing Co., Buffalo, N. Y. Supplement to Engineers' Handbook, 70 pages, 5¼ x 7½ in. Compiled as a supplement to the Lumen Handbook, this data book details the physical and metallurgical properties of the latest non-ferrous alloys.

CERROMATRIX—Cerro de Pasco Copper Corp., 44 Wall St., New York, N. Y. Catalog 36 pages, 8½ x 11 in. Uses and methods of applying Cerromatrix, a low-temperature-melting alloy that expands slightly on solidification, are described and illustrated.

DURALOY—The Duraloy Co., Scottsdale, Pa. Bulletin 3926-G, 8 pages, 8½ x 11 in. Sets forth properties and typical uses of Duraloy alloys for resisting corrosion, high temperatures, and abrasion.

NON-FERROUS ALLOYS—P. R. Mallory & Co., Inc., Indianapolis, Ind. Bulletin M-250, 16 pages, 8½ x 11 in. Physical and electrical properties of Mallory non-ferrous alloys are listed in this bulletin, which also gives application data.

Mechanical Parts

BLOWERS — Roots-Connersville Blower Corp., Connersville, Ind. Bulletin 32-33-B-11, 20 pages, 8½ x 11 in. Illustrates and describes rotary positive gas exhausters and boosters, giving pressure-volume curves and information regarding operating characteristics of this equipment.

BRONZE BEARINGS—Johnson Bronze Co., New Castle, Pa. Slide rule chart provides, in a form convenient for engineers, designers and draftsmen, size information on over 800 sizes of standard stock bronze bearings.

HYDRAULIC COUPLING — Hydraulic Coupling Div., American Blower Corp., 6000 Russell St., Detroit, Mich. Bulletin 3719, 8 pages, 8½ x 11 in. The Scoop Control Fluid Coupling for industrial variable speed control is described and illustrated in this bulletin. Engineering drawings show operation in detail.

LUBRICATION—Bijur Automatic Lubricating Corp., Long Island City, N. Y. Bulletin J, 3¾ x 9 in. Discusses automatic lubrication, and lists 14 reasons for its use.

PUMPS—John S. Barnes Corp., 301 So. Water St., Rockford, Ill. Two bulletins, 4 pages each, 8½ x 11 in., illustrate, describe and give performance data on Barnes gear pumps and piston pumps.

ROLLER BEARINGS—Orange Roller Bearing Co., Inc., Orange, N. J. Engineering Data, 16 pages, 8½ x 11 in. Contains complete engineering data, specification tables, and engineering diagrams of Orange staggered roller bearings.

SOCKET SCREWS—Bristol Co., Waterbury, Conn. Catalog describes features of the multiple spline socket screw. Contains prices and complete specification data of Bristo socket set screws and cap screws.

SPEED CONTROL—Reeves Pulley Co., Dept. A., Columbus, Ind. Catalog G-397, 122 pages. Complete information on the entire Reeves Line of speed controls is given in this new revised catalog which is profusely illustrated with installation views.

SPEED REDUCERS — Philadelphia Gear Works, Venango & G Sts., Philadelphia, Pa. Catalog H-39, 64 pages, 8½ x 11 in. Covers single, double and triple reduction herringbone speed reducing units. Couplings and a number of other products are also illustrated and described in detail.

TRANSMISSION EQUIPMENT—The Medart Co., 3500 DeKalb St., St. Louis, Mo. Catalog 66, 144 pages, 8½ x 11 in. This catalog gives complete engineering data and list prices for all types of Medart power transmission equipment. Cross-indexed for quick reference.

WASHERS—Wrought Washer Mfg. Co., 2100 S. Bay St., Milwaukee, Wis. Catalog 55-C. This new publication lists thousands of washer specifications in various materials, including steel, brass, copper, aluminum, fibre, carried in inventory.

Electrical Parts

ELECTRICAL EQUIPMENT — Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa. Catalog 30-000, 59 pages, 8½ x 11 in. Designed to help the user of electrical apparatus, this "Quick Selector" catalog contains application data, circuit diagrams and tables describing all types of motor or lighting circuit equipment.

GEAR MOTORS—The Holtzer-Cabot Electric Co., Boston, Mass. Bulletin 1480, 4 pages, 8½ x 11 in. Describes and gives technical data for the new type RWC-25 precision capacitor type synchronous motors.

MAGNETIC STARTERS—General Electric Co., Schenectady, N. Y. Bulletin GEA-3250, 4 pages, 8 x 10½ in. Describes the new smaller size 0 starter. Bulletin GEA-3248, 4 pages, 8 x 10½ in. Describes and sets forth design features of the new d.c. magnetic starter, type CR 4052.

MOTORS—Emerson Electric Mfg. Co., St. Louis, Mo. Folder X-3558, 4 pages, 8½ x 11 in. Illustrates and briefly describes the complete Emerson line of stock motors, 5 hp. and smaller.

RELAYS—Struthers Dunn, Inc., 1315 Cherry St., Philadelphia, Pa. Catalog, 32 pages, 8½ x 11 in. Describes many new items, and contains information on relays, timing devices and thermostats.

SYNCHRONOUS MOTORS—General Electric Co., Schenectady, N. Y. Bulletin GEA-1191B, 52 pages, 8¼ x 10¾ in. This booklet is a general treatise on synchronous motors, discussing their economies, general description, construction and operating characteristics. Many illustrations show applications.

TIME SWITCHES—General Electric Co., Schenectady, N. Y. Bulletin GEA-3339, 8 pages, 8 x 10½ in. Describes and gives specifications for Type T-27 general-purpose automatic time switches. Dimensions and connection diagrams are included.

Fabrication Methods

CONTOUR SAWING—Continental Machines, Inc., 1301 Washington Ave. South, Minneapolis, Minn. Handbook, 160 pages, 6¼ x 9¼ in. This complete treatise gives detailed information relating to the operation and application of contour sawing. Newly compiled data relating to selection and cutting speeds of saws and band files is included.

WELDING—General Electric Co., Schenectady, N. Y. Bulletin GEA-1546-F, 40 pages, 8 x 10½ in. Prepared as a guide for the proper selection of electrodes, this publication gives important suggestions on welding technique, characteristics of deposited weld metal, charts of joint forms and positions, and a handy estimator for electrode quantities needed to complete a job.

Finishes

FINISHING METHODS—Maas & Waldstein Co., 438 Riverside Ave., Newark, N. J. "A Review of Finishing Methods," 6 pages, 8½ x 11 in. A comprehensive review of various methods of applying lacquers and enamels to metal products.

FLAME CLEANING—Air Reduction Sales Co., 60 East 42nd St., New York, N. Y. "Flame Cleaning and Dehydrating," 8 pages, 8½ x 11 in. This illustrated booklet discusses the use of flame cleaning and dehydrating to halt deterioration of old steel structures.

LITHOFORM—American Chemical Paint Co., Ambler, Pa. Bulletin 7-2-3, 4 pages, 8½ x 11 in. The nature and use of Lithoform in giving good paint adhesion characteristics to galvanized iron, zinc and cadmium surfaces are explained in this bulletin.

WRINKLE FINISHES — New Wrinkle, Inc., Dayton, Ohio. "Do You Finish Castings?" 4 pages, 8½ x 11 in. Sets forth the advantages of the recently developed New Wrinkle dip-process of casting finishing.

Miscellaneous

ANTI-FREEZE SYSTEM—Sullivan Machinery Co., Michigan City, Ind. Bulletin R-16, 4 pages, 8½ x 11 in. Describes anti-freeze systems for air lines: Tannergas for temporary air compressor installations, and Frosto for permanent installations where electric current is available. The newly developed Frosto Vaporizer for portable compressors is also described.

RADIUM DETECTOR—Geophysical Instrument Co., 1315 Half St. S.E., Washington, D. C. Circular "E", 4 pages, 8½ x 11 in. Illustrates and describes the new radium detector for discovery of lost radium, measurement of radioactive preparations and investigation in radioactivity.

Magnet Coils

Proportioning and Testing

GRAHAM LEE MOSES
Westinghouse Electric & Manufacturing Company

IN calculating the proportions of a magnet coil, the size of the core on which the coil operates is usually determined by the magnetic requirements of the device. Therefore, there is usually a minimum inside diameter for the coil winding. The designer has control only of the length and wall height of the winding and often these have maximum limits. In selecting the proper proportions of a magnet coil the fundamental relations must be studied.

When the size of the tube on which the coil is to be wound has been established and the maximum outside diameter of the coil is fixed, it is often desirable to vary the length to establish a proper balance between the watts generated within the winding and the ability of the coil to dissipate this heat. For a specific voltage and approximate mean diameter of winding, the ampere-turn requirements fix the size of wire (PRODUCT ENGINEERING Reference Book Sheet, page 503, December, 1938). For this particular size of wire the total watts generated decrease as the resistance increases, for if $P = E^2/R$, then P varies inversely as R. As the resistance for a coil with a particular size of wire varies with the volume of the winding, the volume determines watts generated.

If the inside and outside diameters of the winding remain relatively constant, the barrel surface of the coil changes in the same ratio as the volume when the length is increased or decreased. Since increasing the length of a coil decreases the heat generated within and increases the coil's ability to dissipate this heat, it is an important factor in determining the ultimate stabilized temperature rise the coil attains.

The barrel surface of a coil per inch of length is equal to its outside diameter multiplied by π (3.1416). Its outside diameter can be expressed as:

$$A/L = \pi(d + 2w).$$

In the same manner the volume of the winding per inch length is equal to its mean circumference multiplied by its wall thickness. As its mean diameter is equal to the sum of the outside diameter of the tube on which it is wound and its wall thickness this can be expressed as $V/L = \pi\ (d + w)\ w$.

For a given size of wire, as established by the ampere-turn requirements, the volume of the winding determines the coil resistance:

$$R = Vr = [\pi(d + w)w]\ Lr.$$

Since the barrel surface area determines the watts which a coil can dissipate at a given temperature rise, the relation between voltage, resistance per inch length and watts per inch length can be compared to barrel surface area as follows to express permissible watts dissipation per unit length of coil.

$$\frac{E^2}{R} + L = \frac{P}{L} = \frac{A\,k}{L} = \pi\ (d + 2w)\ k$$

By combining equations, a single formula can be derived for the length of winding to balance watts generated and watts dissipated:

$$\frac{E^2}{R} = L \times \frac{A}{L} \times k = L\,[\pi\ (d + 2w)\ k]$$

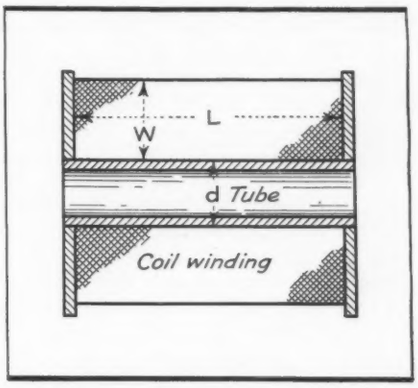
$$L = \frac{E^2}{R \times \pi\ (d + 2w)\ k}$$

$$L = \frac{E^2}{[\pi\ (d + w)\ wLr]\ [\pi\ (d + 2w)\ k]}$$

$$L^2 = \frac{E^2}{\pi^2\ [(d + w)\ rw]\ [(d + 2w)\ k]}$$

$$L = \frac{E}{\pi}\sqrt{\frac{1}{wrk\ (d + w)\ (d + 2w)}}$$

Since it is not so simple to solve these formulas for w at a balanced condition they have been reduced to express the resistivity constant of the winding. This in turn can be translated into A.W.G. size by curve Fig. 4, page 486 of PRODUCT ENGINEERING, December, 1938.



In approaching the design of a coil having wall height, w, as a variable, a different method of attack must be used. It is necessary to cut and try somewhat in balancing w and wire size for desired ampere-turns against the w and wire size to give balanced heating.

The required area to dissipate the generated watts will be:

$$A = \frac{E^2}{Rk} = \frac{E^2}{rk\,\pi\,wL\ (d + w)}$$

The actual area as determined by physical dimensions is expressed as

$$A = L\,\pi\ (d + 2w)$$

By combining these and reducing, the following expression is obtained for r in terms of the physical, electrical and thermal limitations and requirements.

$$\frac{E^2}{rk\,\pi\,wL\ (d + w)} = L\,\pi\ (d + 2w)$$

Which reduces to

$$r = \frac{E^2}{L^2\,\pi^2\,kw\ (d + w)\ (d + 2w)}$$

General Testing

As a final step, every design no matter how carefully estimated must prove itself capable of proper operation by test. While it is possible to accurately calculate and predict operation of electrical apparatus, if all the important factors are given proper consideration by the designer, occasionally a new factor creeps in or an important one is overlooked. Tests therefore serve the doubly useful purpose of checking the designer's art and the shop's performance in producing the design according to specifications.

Nomenclature

d = Diameter of tube.
w = Wall height of winding.
L = Length of coil in in.
A = Area barrel surface in sq. in.
V = Volume of winding in cubic in.
P = Total watts that can be dissipated from given coil for specified temperature rise.
E = Continuous or R.M.S. voltage for specified temperature rise.
R = Resistance of coil in ohms.
r = Specific resistivity of wire in ohms per cu. in. of winding.
k = Heating constant of coil in terms of watts per sq. in. of barrel surface. For 85 deg. C. rise by resistance 1.35 watts per sq. in. (based on cold resistance) is a good average value.

Magnet coil tests should determine:

1. Sufficiency of insulation
   (a) From winding to ground
   (b) Between turns and layers
   (c) Between start and finish leads
2. Continuity of circuit
3. Accuracy of number of turns
4. Correctness of size of wire
5. Thermal characteristics
6. Operating characteristics

RESISTANCE TESTS are the simplest and yet most fundamental. They should be made on every coil manufactured. For normal production a limit of plus or minus 5 per cent should be allowed over the calculated value. If the resistance exceeds the maximum, the wire size may be small or the coil outside diameter may be too great. If the resistance is too low it is probable that turns or layers are short circuited, or the wire size may be running too large. Where resistance values on several coils are within the calculated limits, it is considered safe to assume that the number of turns and wire size are correct. If a coil passes the resistance test there can be no question of circuit continuity.

DIELECTRIC TESTS. The ground insulation should be tested by an application of overvoltage to insure its sufficiency. These tests should be made after the coil is assembled in place on the apparatus unless it is wound on a metallic spool. The A.I.E.E. specifies the following standards:

VOLTAGE CLASS	ONE MINUTE TEST VOLTAGE
125 volts or less	800 volts
125 to 600 volts	Twice rated voltage plus 1000 volts
600 volts or more	2¼ times rated voltage plus 2000 volts

Alternating current of sine wave form whose r.m.s. value equals test voltage specified is used in this test.

On coils for high voltage duty (300 volts or more) with a large number of turns (10,000 or more) it is desirable to test the internal insulation between turns, between layers and between start and finish leads. This can be done with a standard high voltage test set. The recommended procedure is to first insert a steel core in the coil. Then apply a high a.c. voltage from the test box directly to the coil terminals. The large number of turns around the steel core produces sufficient impedance to prevent tripping the breaker on the test set and yet it applies this test voltage directly to the winding. It is customary to apply the same test voltage between terminals as between winding and ground. However, on coils subjected to high surge voltages it may be desirable to increase this test voltage.

SHORT CIRCUIT TESTING. In some instances undesirable short circuits between coils can be tolerated on d.c. magnet coils, but they must never be permitted on a.c. magnet coils. A coil whose resistance is well up to the average calculated value is usually found to be reasonably free from short circuits. This is not a very accurate check and it is generally necessary to use a special test rig to check short circuits. This takes the form of an "E" shaped high frequency magnet with a balanced wattmeter indication. When the coil under test is placed over one leg of the magnet, short circuits therein act as a short circuited transformer secondary causing an unbalance on the meter element which indicates degree of short circuit.

OPERATING TESTS. Upon the completion of a new design it should be given a complete functional test to check the effectiveness of the electro-magnetic features. Due consideration must be given to change in resistance of the magnet coils due to heating in normal duty. Unless the magnetic devices are subject to very accurate control of tolerances, an allowance of at least 10 per cent should be made over the observed operating characteristics to insure that all devices come within the limits.

Temperature Testing

While operating temperatures can generally be predicted with reasonable accuracy, it is advisable to make a temperature test on at least one coil of each new basic design to insure that the coil is operated within the limits prescribed for its class of insulation.

There are two general methods of measuring temperature on magnet coils, namely, (1) Surface temperature indicators such as thermometers; (2) Change in resistance of winding.

The first necessarily gives an indication that is lower than internal temperatures and the second gives an indication that is an average of the winding temperature. For purposes of standardization the A.I.E.E. has established a differential between the maximum permissible observed temperature rise by these two methods for different classes of insulation. These are based upon an ambient temperature of 40 deg. C.

MAX. PERMISSIBLE RISE	INSULATION CLASS A	INSULATION CLASS B
Thermometer method	65° C.	85° C.
Resistance method	85° C.	105° C.

In the test procedure, the complete apparatus is set up with coil assembled in its normal operating position. Several thermometers or thermocouples are applied to the coil surface so as to obtain an average indication, and a thermometer is placed in room air adjacent to the apparatus to record ambient temperature. Next, accurately measure the coil resistance and record it with the coil surface temperature as well as the ambient temperature.

Only after these preliminaries should power be applied. The voltage selected should be held constant throughout the test. During the early part of the test frequent readings should be taken, but as the coil temperature approaches a steady state, less frequent readings will suffice. The following data should be observed:

Time of reading
Volts applied to coil terminals
Amperes flowing through coil
Surface temperature
Room air temperature

From the volts and amperes observed, an approximation of the coil resistance can be obtained. Then, the coil temperature by resistance change method, T, can be determined by the following formula:

$$T = R/r\ (234.5 + t) - 234.5$$

where

R = Hot resistance of coil, ohms
r = Cold resistance of coil, ohms
t = Temperature of coil at start, deg. C.

After all observed values reach a steady state it is desirable to take a set of "shut down" readings. Arrangements should be made before power is shut off to quickly change the coil connections to a Wheatstone bridge. After power is shut off frequent readings of coil resistance should be made until the change in coil resistance is at least half its total increase due to temperature.

After assembling the test information and the significant calculated data, a time-temperature rise curve and a cooling curve should be constructed. In general the actual observed maximum temperature rise will not be exactly what was estimated at the test voltage, but conclusions as to the reliability of the apparatus can readily be drawn.

Tests are the criteria by which all designs and products must be judged. Therefore it behooves the designer to specify significant and complete tests and the tester to painstakingly perform these tests so as to leave a clean cut record of the true performance of the apparatus. To be of greatest usefulness test results must be recorded, analyzed and used on future jobs. Such recorded experience forms the background for future advances in the art.